

AUTOMOBILE ENGINEER

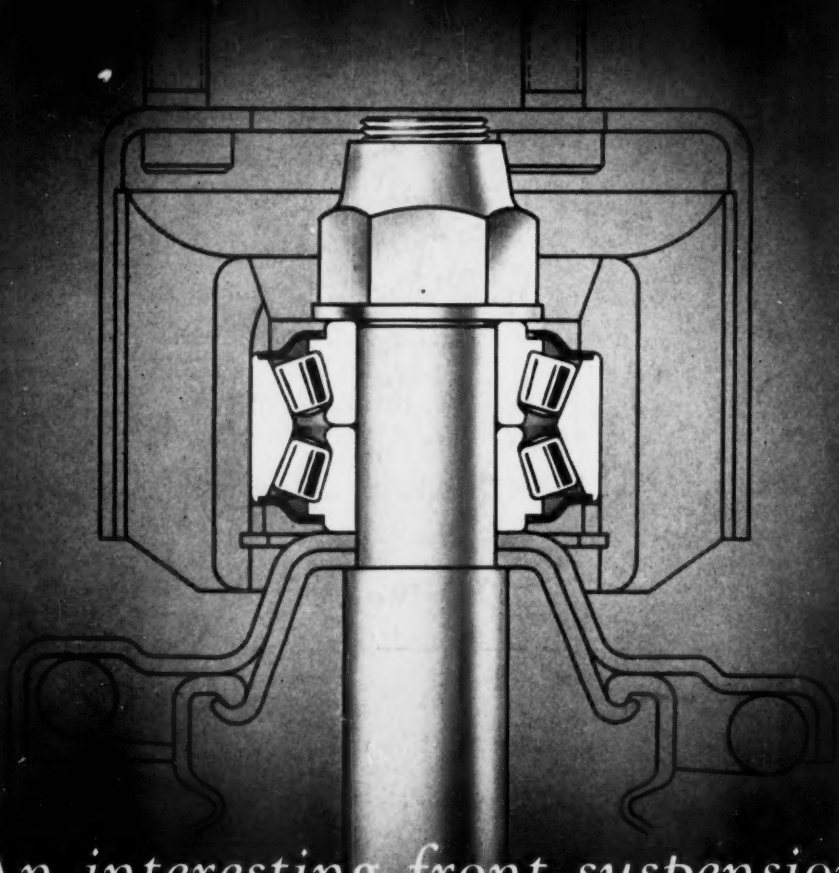
DESIGN · PRODUCTION · MATERIALS

Vol. 43 No. 574

DECEMBER 1953

PRICE: 3s. 6d.

ON THE CONSUL AND ZEPHYR SIX



*An interesting front suspension
with*

TIMKEN

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BRITISH TIMKEN LTD., DUSTON, NORTHAMPTON; AND BIRMINGHAM

CYCLONE

HACKSAW BLADES

THEY KEEP ON CUTTING LONGER

The secret of this lies in the rigid control exercised from the raw material to the finished job—a factor ensuring the production of blades of maximum efficiency.

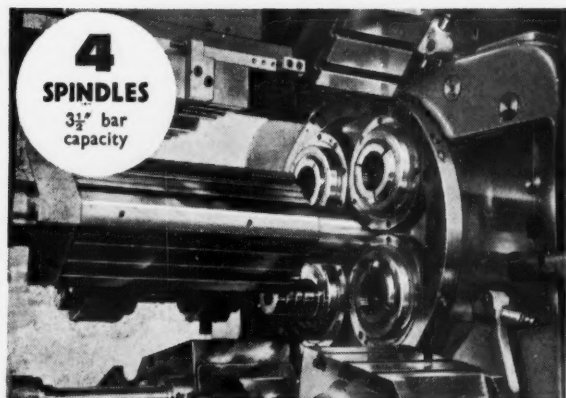
Whatever your needs, we can supply hacksaw blades which will "keep on cutting longer."



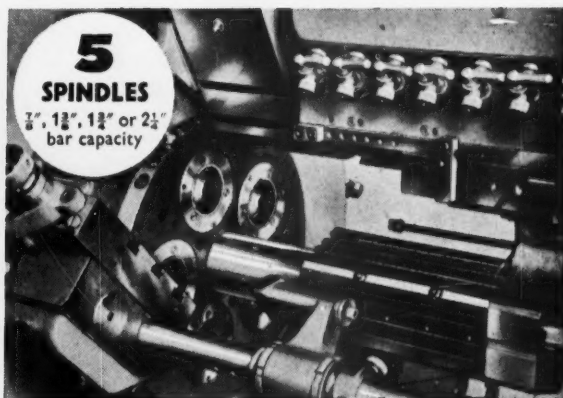
ENGLISH STEEL CORPORATION LTD.
Holme Lane Works, Sheffield



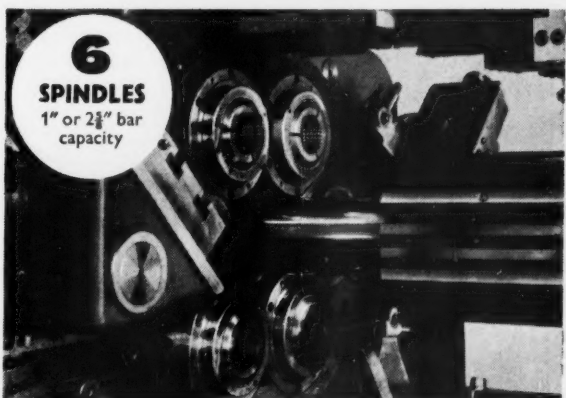
FOUR, FIVE, SIX AND EIGHT SPINDLE AUTOMATICS provide production engineers with new opportunities to increase output and lower costs



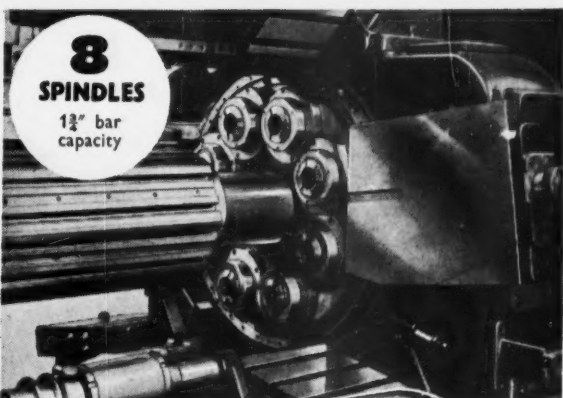
**4
SPINDLES**
3½" bar
capacity



**5
SPINDLES**
1", 1½", 1¾" or 2½"
bar capacity



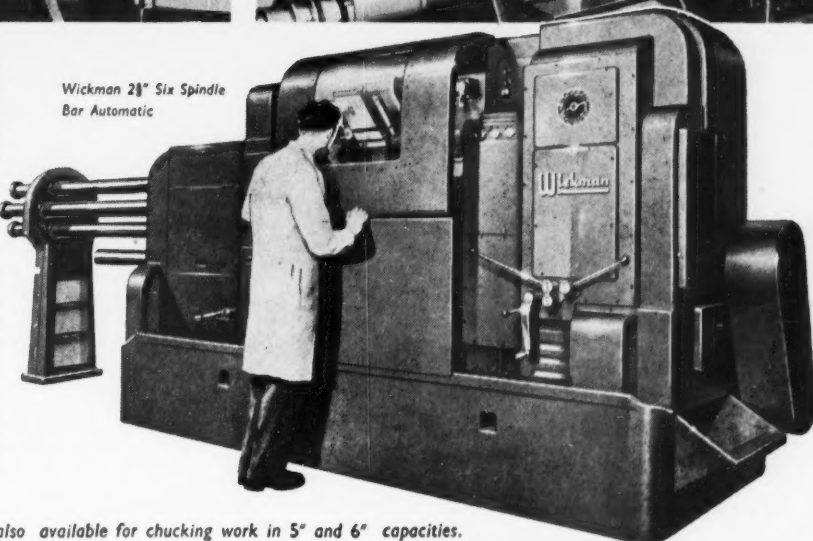
**6
SPINDLES**
1" or 2½" bar
capacity



**8
SPINDLES**
1½" bar
capacity

Fast producing Wickman Multi-spindle Automatics can now be applied to a wider range of jobs than ever before. The tooling opportunities can be easily visualised from the illustrations above. The Wickman patent auto-setting mechanism is incorporated in all machines of the range and alterations to tool feed strokes and bar feed are accomplished without cam changing. The full fast approach stroke is unaltered by this mechanism, setting up is simplified and change-over time reduced—that's why these automatics can be considered for short run jobs as well as long runs on one component.

Wickman 2½" Six Spindle
Bar Automatic



Wickman 5-spindle automatics are also available for chucking work in 5" and 6" capacities.

WICKMAN of COVENTRY

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LEEDS • GLASGOW • NEWCASTLE • BELFAST



280 WM



Rover Car Factory, Solihull. High intensity lighting in a body spray tunnel by fluorescent lamps in a glazed enclosure.

Tailored for the job

The lighting of many processes is vital to the smooth and rapid flow of work and to the quality of the finished product. For example, poor lighting could make a spray tunnel into a bottle-neck — each job taking a little too long, a little portion missed, a return to the spray line — and so the whole production line marks time. Whatever form it takes, good lighting not only helps to provide a satisfactory working environment but is an active production tool.

Fluorescent lighting is as good as daylight — only more consistent. It is efficient; it is economical; and it is *flexible*. You can 'tailor' it, easily and exactly, to the special requirements of production at all stages.

Electricity for PRODUCTIVITY

HOW TO GET MORE INFORMATION

Your Electricity Board will be glad to advise you on how to use electricity to greater advantage — to save time, money, and materials. The new Electricity and Productivity series of books includes one on lighting — "Lighting in Industry". Copies can be obtained, price 9/- post free, from E.D.A., 2 Savoy Hill, London, W.C.2, or from your Area Electricity Board.

Issued by the British Electrical Development Association



TOUCHÉ !

"On the wrong foot again, old boy.

When will you learn . . . !"

"When I've been at it as long as you —

I might make a better showing."

"Don't sound so glum —

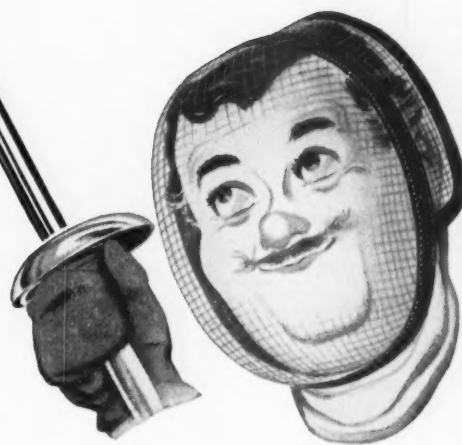
you've taught me one thing—

and I'm your humble servant."

"Ah! back to M. & C. Service —

my 'coup de grace'."

"Never a moment's worry! On your Guard!"



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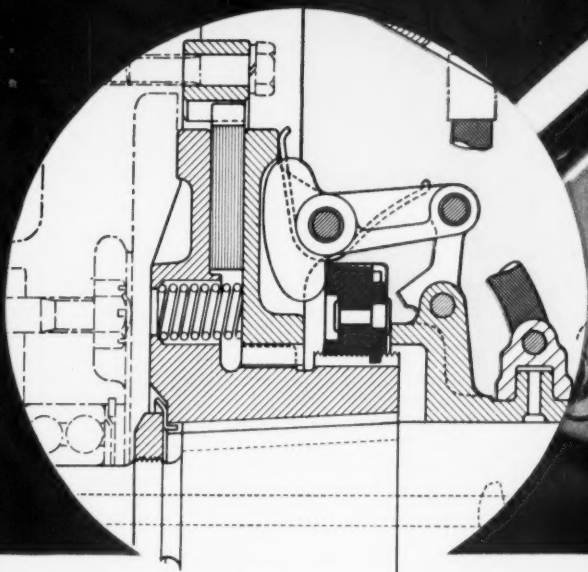
The logo for Bury Felt is presented within a rectangular frame. The word "BURY" is in a large, bold, black, sans-serif font on a white background. The word "FELT" is in a large, bold, white, sans-serif font on a dark, textured background.

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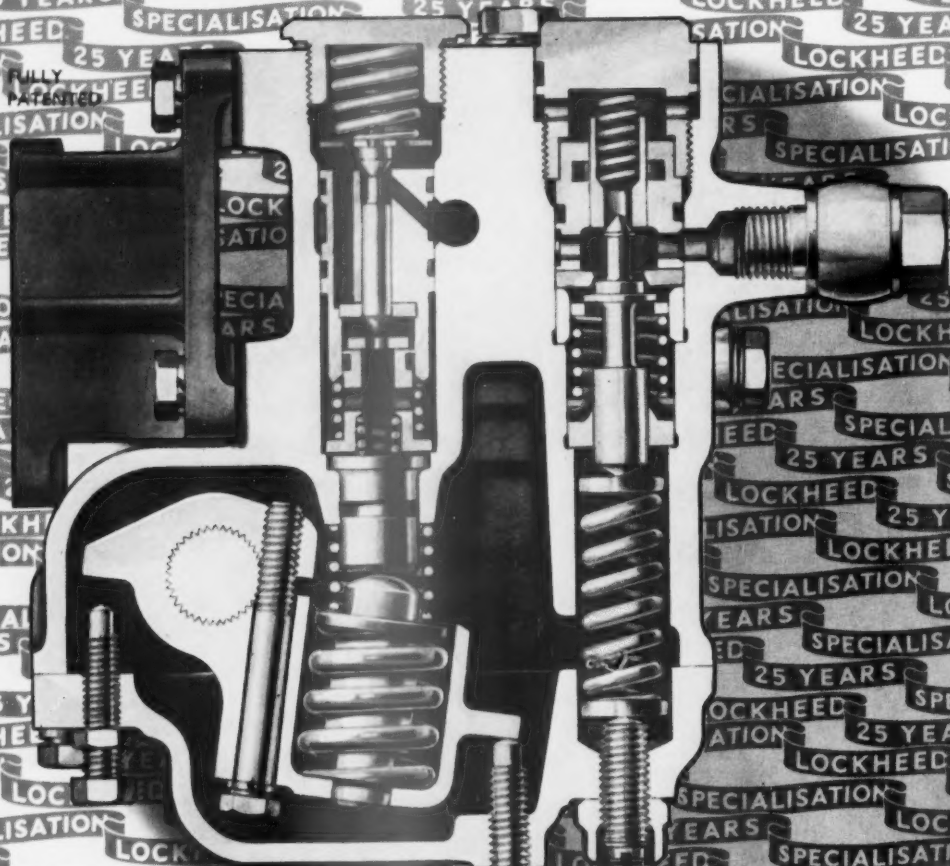
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ROCKFORD
POWER TAKE OFFS & CLUTCHES



FULLY
PATENTED



LOCKHEED

REGD TRADE MARK

THE HEART OF THE SERVO

The power valve system

In this type of Lockheed Servo system, used on trolley buses, and elsewhere, where it is desired to have an extremely light brake-pedal pressure, a master cylinder is not employed, the fluid for brake operation being derived direct from the pump or pressure reservoir.

In the unit illustrated, the cut-out valve and power valve are combined.

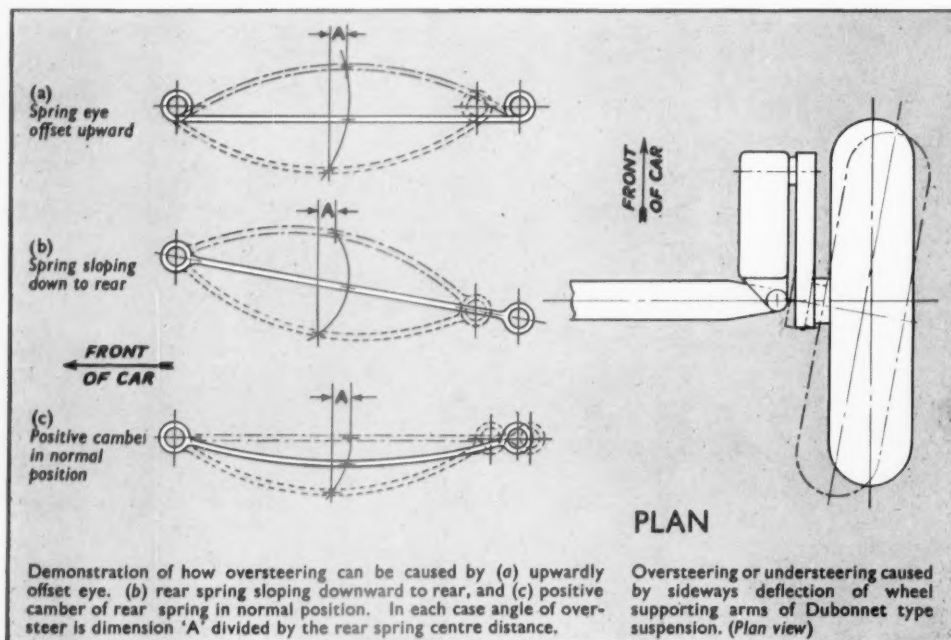
The system balances the brake-line pressure against the pedal pressure, giving normal feeling of control.

When subsidiary services, such as power-assisted steering, door-opening equipment etc., are added, a secondary pressure reservoir is employed, in conjunction with a non-return valve leaving the primary pressure reservoir for brake operation only.

AUTOMOTIVE PRODUCTS COMPANY LIMITED, LEAMINGTON SPA, ENGLAND

SUBTLETIES OF STEERING

Causes of over- and under-steer: 6. Car properties (c) 'Roll steer'



WE must not forget, in considering the car properties which affect its behaviour in over- and under-steer, that there are direct influences called 'roll steer' as well as the indirect influences which we have so far been considering and which act through the tyre properties.

'Roll steer' can exist at both front and rear of a car. In its simplest and most obvious form, it is the movement of a beam axle in plan view by some influence which acts as the car rolls on a corner as a result of sideways acceleration and the corresponding forces. The semi-elliptic leaf spring, in locating the axle, behaves like a link whose effective length is about three quarters of its half length. If these links (one on each side of the car) are parallel with the ground in their initial position, then rolling of the car will move each end of the axle forwards or backwards by the same amount and there is no 'roll steer'. This happy condition can only exist at one state of load of the axle concerned. If the springs are fixed at their front ends and if the effective links slope downwards towards the axle; then roll will move the outer end of the axle backwards in relation to the inner (Bastow, "Steering Problems & Layout", Proc. I.A.E. 1937-8) and will in effect increase the drift angle for that pair of wheels by this steering angle of the axle.

'Roll steer' can also occur on a front axle if the geometry of the steering drag link is not perfect (and may in that case differ between right hand and left hand corners), and exists with independent front suspension if the geometry of the steering connections is imperfect in such a way that steering

angle occurs on bump and rebound, and if the amount of 'toe-in' for equal amounts of bump and rebound is not the same.

Undue initial 'toe-in' on the front wheels gives an over-steer tendency, because as the weight is transferred to the outer wheel on a corner it tends to take charge and therefore reduce the effective front drift angle.

There are other possible ways of causing 'roll steer', for instance by elastic deflection of parts when subjected to the sideways loads of cornering, but we need not consider every such possibility in detail. Readers can no doubt work out their own possibilities or actualities.

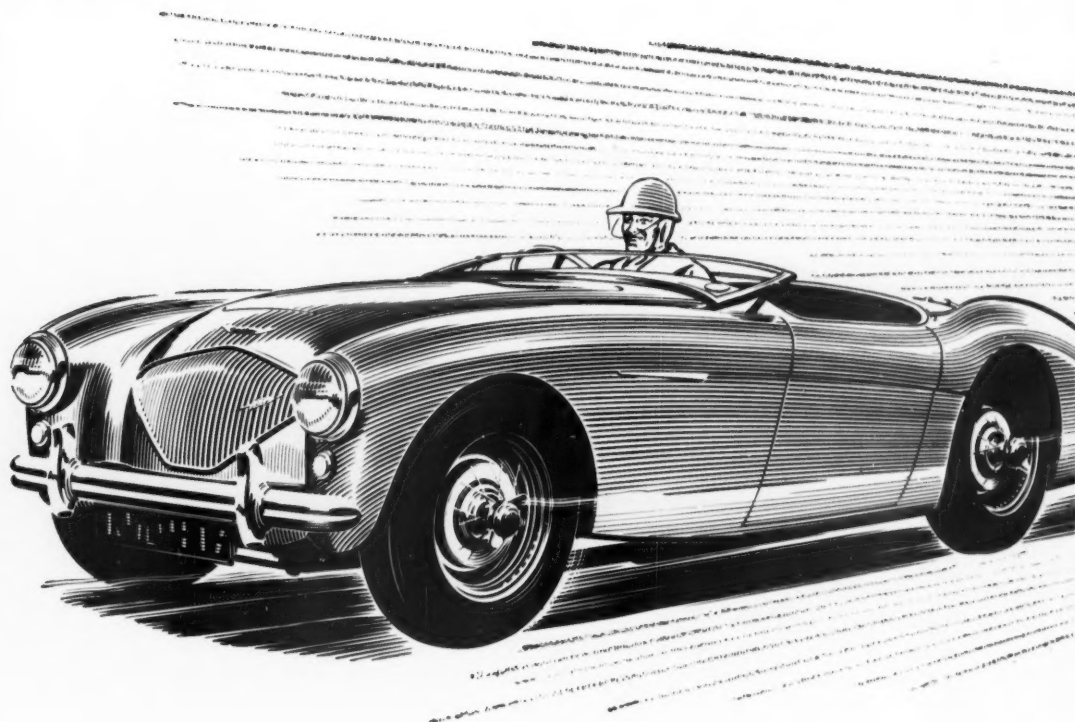
In general, excessive resultant 'roll steer' is to be avoided, because it tends to peculiarities of behaviour in the transient state at the beginning and end of a corner, and because it implies differences in behaviour between the unrolled and rolled car. If under-steer is obtained only by 'roll steer' effects, then it is not present when a car is hit by a gust of wind, and one of the reasons for liking under-steer is the stability it provides in gusty conditions.

Where a rear axle is located by semi-elliptic springs (Hotchkiss drive) it does provide some measure of correction for increasing load at the rear of the car, since reduction of standing height at the rear gives an under-steer tendency, whereas the extra weight on the tyres which causes this reduction in standing height will, as we have seen, increase the over-steer tendency.

Thompson
Self-adjusting

STEERING ROD ASSEMBLY

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The world record-breaking Austin-Healey 100

car was equipped with bodywork entirely fabricated

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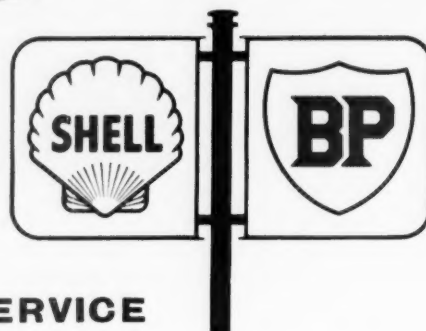
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Glad to be of service, Mr. Rimmer*

As you are no doubt aware, we have now been in business at the above address for the past 18 months and in the first 12 months more than doubled the previous owners' petrol and oil sales. During the first 6 months our total petrol sales were 32,000 gallons, and during the first 6 months of our second year our total sales are over 40,000, showing a sales increase of over 8,000 gallons, together with satisfactory increases in our oils sales. We should like to express our very great thanks for the kindness, co-operation, and assistance which your organisation has always shown towards us, and without which on occasion we should have been at a loss.

* Mr. Rimmer writes on behalf of Riddington and Rimmer (Birkdale Garages) Ltd. Southport.



THE SIGN OF FRIENDLY SERVICE

"The castings are characterised by dimensional accuracy, good surface finish, soundness and freedom from hard spots"



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We choose Harper castings firstly for quality and secondly for reliability of delivery. We have dealt with Harpers for many years, and we consider that they have contributed to a considerable extent to maintaining the high reputation of Villiers engines, which are sold in very large quantities throughout the world".

Harper quality covers iron castings, and also metal pressings, machining, enamelling and other finishes and sub-assembly work.

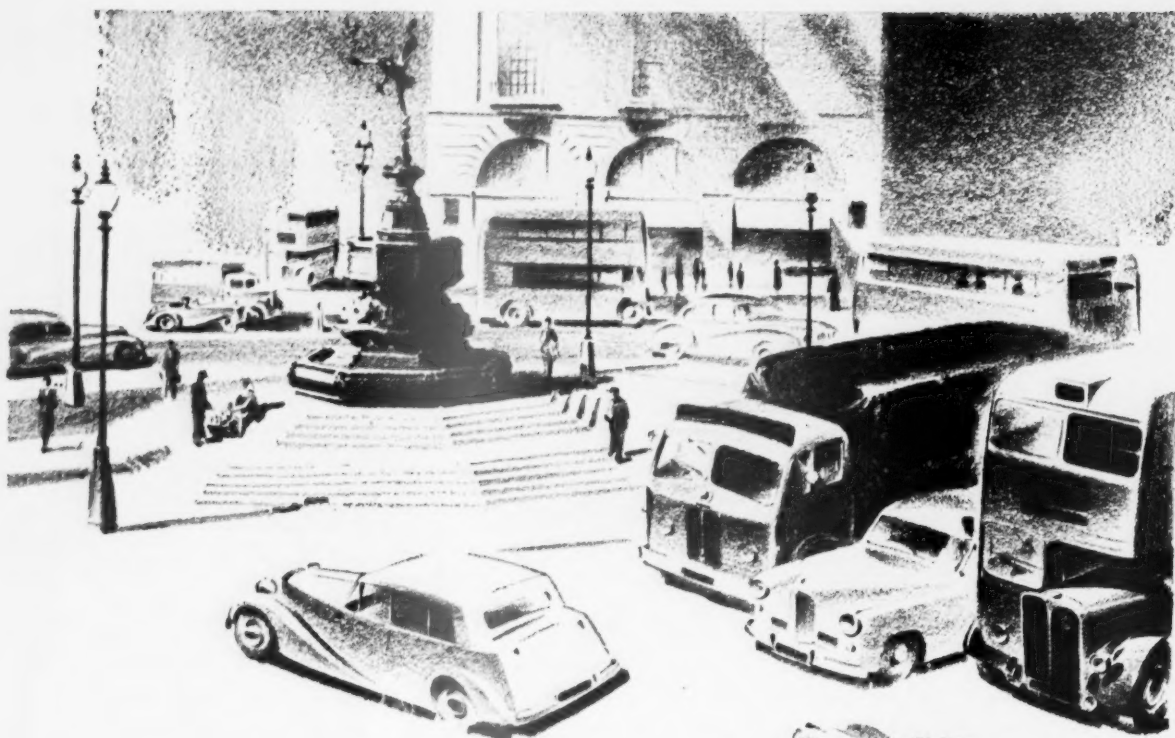
HARPER CASTINGS



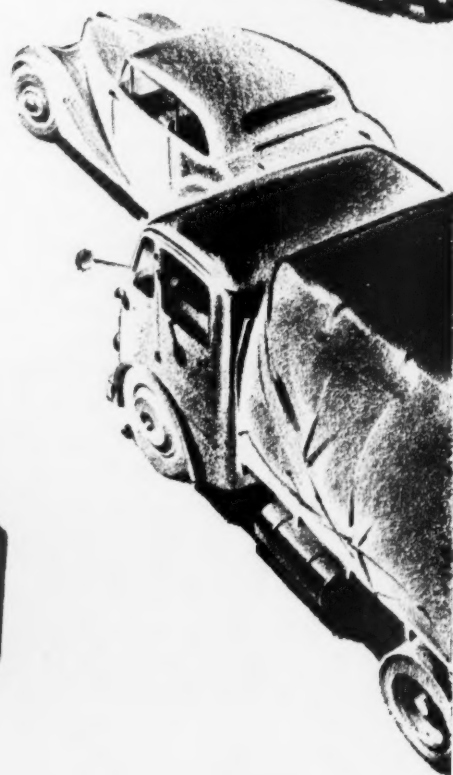
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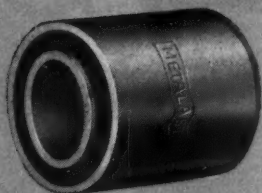


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ON THE FORD

Consul & Zephyr



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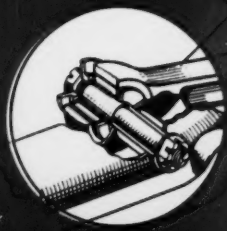


ALSO TWO PAIRS OF BONDED-CONE BUSHES



AND TWO PAIRS OF BONDED-CONE TUBE BUSHES

I·F·S has many variants but few have departed so boldly from the now-conventional wishbone as the Ford design. This takes the fullest advantage of modern rubber engineering and utilizes Metalastik Ultra - Duty bushes and Bonded-Cone bushes. These, with a sustained behaviour unequalled by any other bushes, are used at all three critical points controlling the front-end geometry, resulting in handling properties that evoke warm praise.

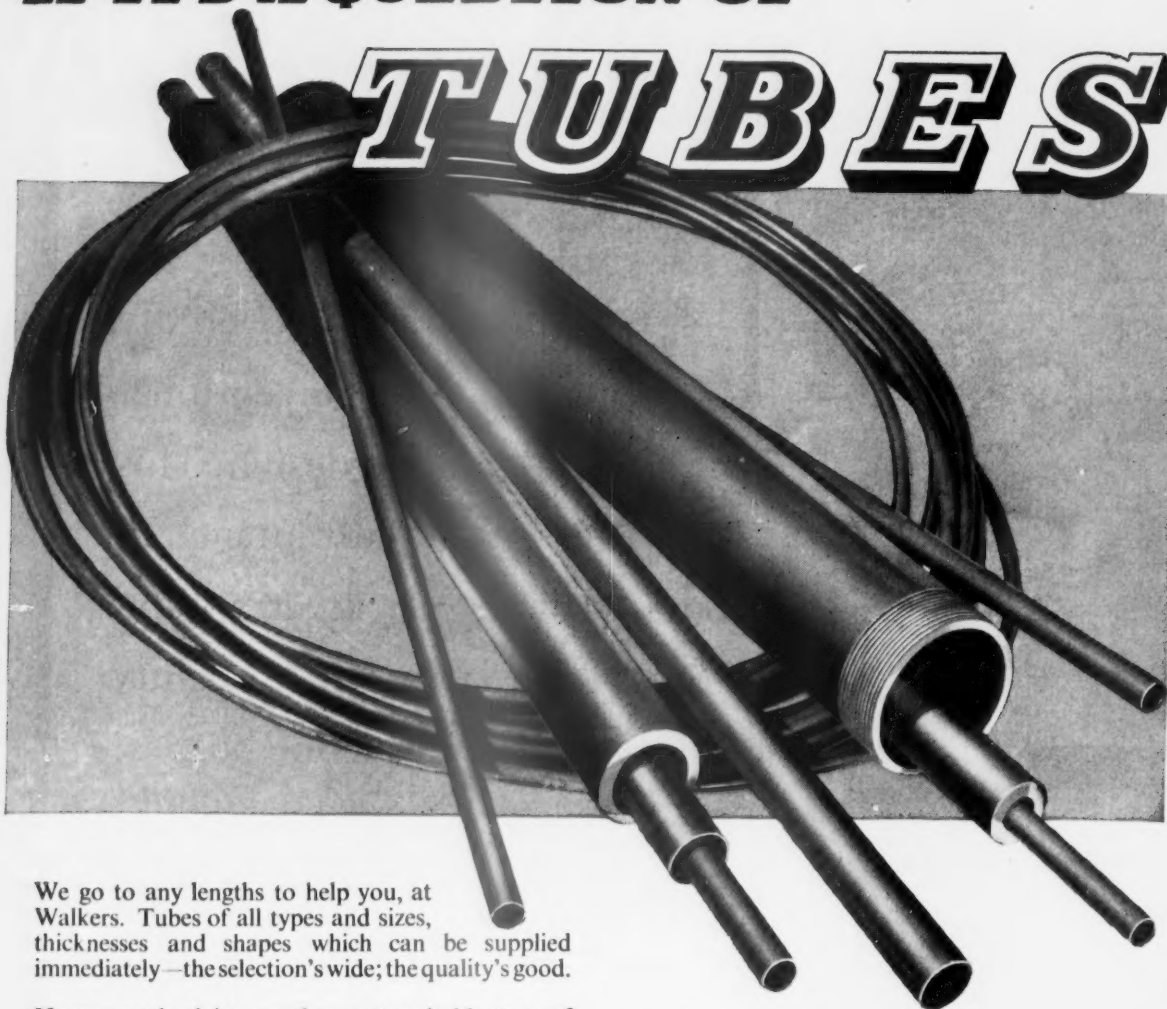


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The use of 'O' rings on reciprocating pistons

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A wide range of sizes are available and we shall be pleased to receive details of your specific requirements.

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10% EXTRA OUT OF THE BLUE



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GEAR BOX AND CLUTCH
HOUSING DIE CAST
IN ALUMINIUM ALLOY



CYLINDER BLOCK IN
"STERLING" CAST IRON



Illustrations by courtesy of The
Standard Motor Company Ltd.



by

STERLING METALS LTD.

Coventry

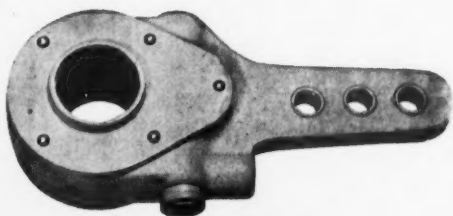
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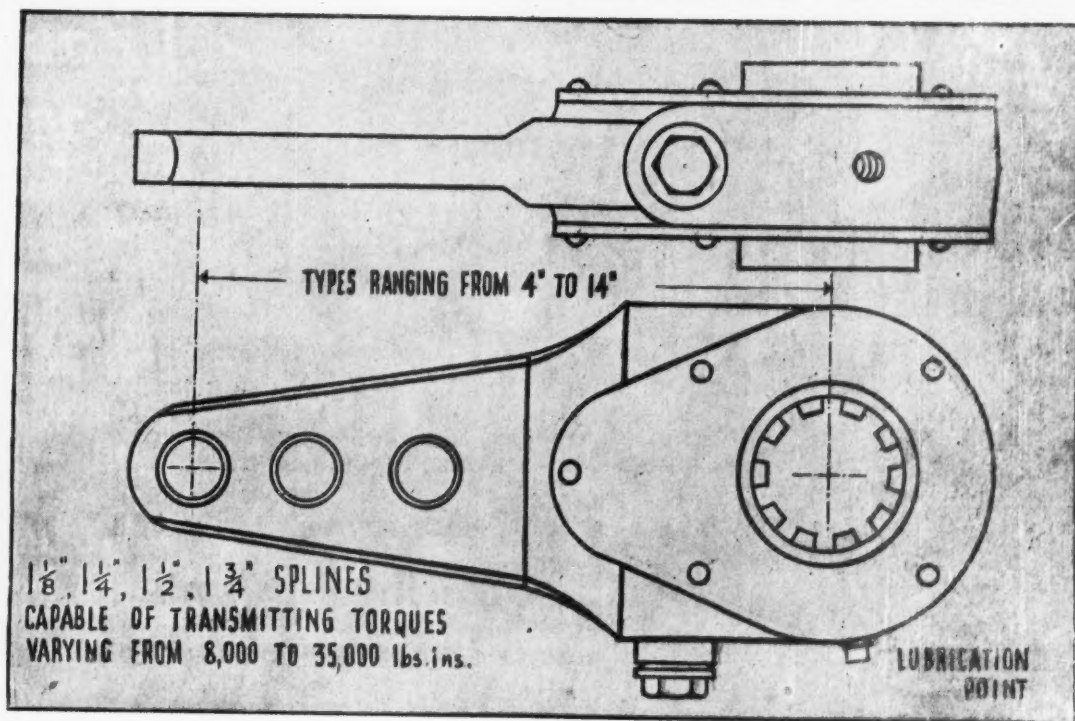
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NEW TYPE SLACK ADJUSTER

which provides most accurate brake adjustment with speed and ease. Made in varying styles and arm lengths suitable for cam operated brakes. Strictly interchangeable in detail with the world renowned



Bendix-Westinghouse SLACK ADJUSTER



CLAYTON DEWANDRE CO. LTD. *Lincoln*

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As competition returns to the motor industry's home market, designers of mass-produced cars will start thinking seriously about aluminium for coachwork. From the beginning it has been used for the bodywork of luxury cars and sports models, and its durability is proved by the excellent condition of the many old aluminium-bodied cars still on the road. In recent years considerable experience has been gained with aluminium body pressings and manufacturers will be able

to put this to good account when discerning motorists are again able to choose freely.

We shall be glad to give, without obligation, specific advice on any application of Noral alloys.

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I don't give two hoots for your Owl Patrol, said the M.D. testily. My little lot have won more badges for drilling in odd corners, screwing up the inscrutable, nut running, gun running, rum running and showing old ladies under buses . . . Good turns? Me newest tenderfoot does them at the rate of 1,000 a minute.* I am happy, nay proud, to be their dear old scout master —

'Old master is right', said a little horror in very long shorts. At least that's what we thought he said.

* The old boy seems to be referring to the latest Desoutter straight drill, a $\frac{1}{2}$ " model weighing only half as much as other tools of similar capacity, with speeds of 1,000, 550 and 370 r.p.m.

'Morse, of course', said the Little Horse.

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**GEAR LAPPING
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.....headline in Production
Engineering & Management
January, 1953

"FORD HOBS GEARS 350% FASTER"

We've checked on it and the editors of PRODUCTION ENGINEERING & MANAGEMENT are right. At Ford Motor Company, U.S.A. 3 $\frac{1}{4}$ " diameter, 9-pitch, 1 $\frac{3}{8}$ " face width transmission gears are being hobbled at a production rate 350% greater per spindle than compared with multiple spindle machines. To do this Ford uses one of the new Michigan Ultra-Speed gear hobbers equipped with double thread accurate unground Michigan hobs. Gears are hobbled two at a time, 58 seconds per pair.

Despite the high output rate, 225 gears are produced per sharpening of the hob. Among the reasons given are that "the machine is of rugged, compact construction, simplified in design, with few gears for the index and main drive and . . . with a maximum spindle speed of 1000 r.p.m."

Of interest is that better control of surface finish is obtained by easier subsequent shaving. Loading and unloading time is kept to a minimum by hydraulically-actuated expanding arbors. Vibration has been practically eliminated by providing maximum rigidity plus a flywheel to dampen torsional vibration.

For further details, write for Bulletin #1458-52.

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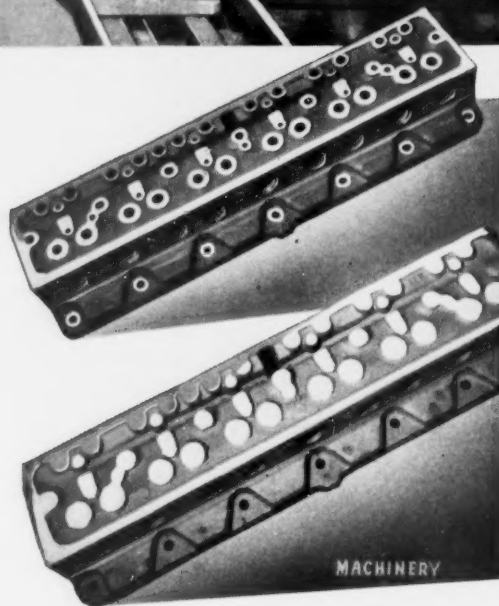
ON "BEDFORD" CYLINDER
HEADS VAUXHALL USE . . .

ARCHDALE *Specials*

By courtesy of VAUXHALL MOTORS LTD., LUTON, we illustrate a special ARCHDALE machine for performing drilling, reaming, counter-boring and tapping operations on cylinder heads in an automatic, continuous sequence.

Developing such machines is a speciality of the ARCHDALE organisation, and the machine illustrated is only one of hundreds of special machines, including transfer machines, supplied to the automobile industry.

Our experienced technical staff will be pleased to co-operate with you on any problems connected with drilling, milling or allied operations.



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Moulded Camshafts

PAR EXCELLENCE

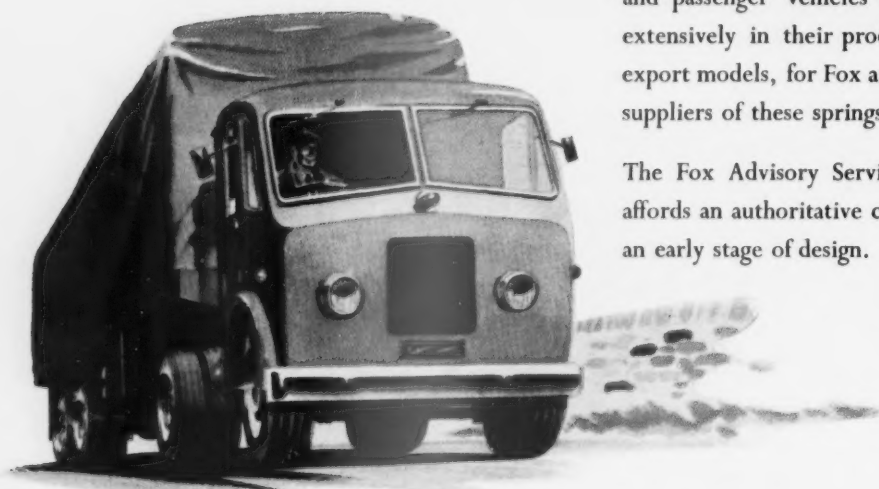
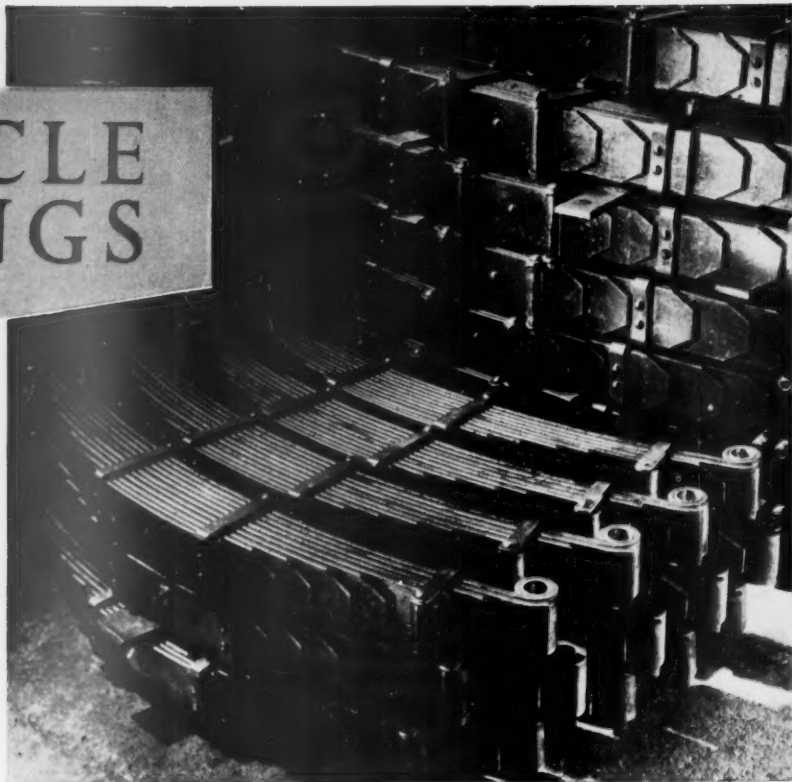


Cast in
MONIKROM
the perfect Camshaft Material



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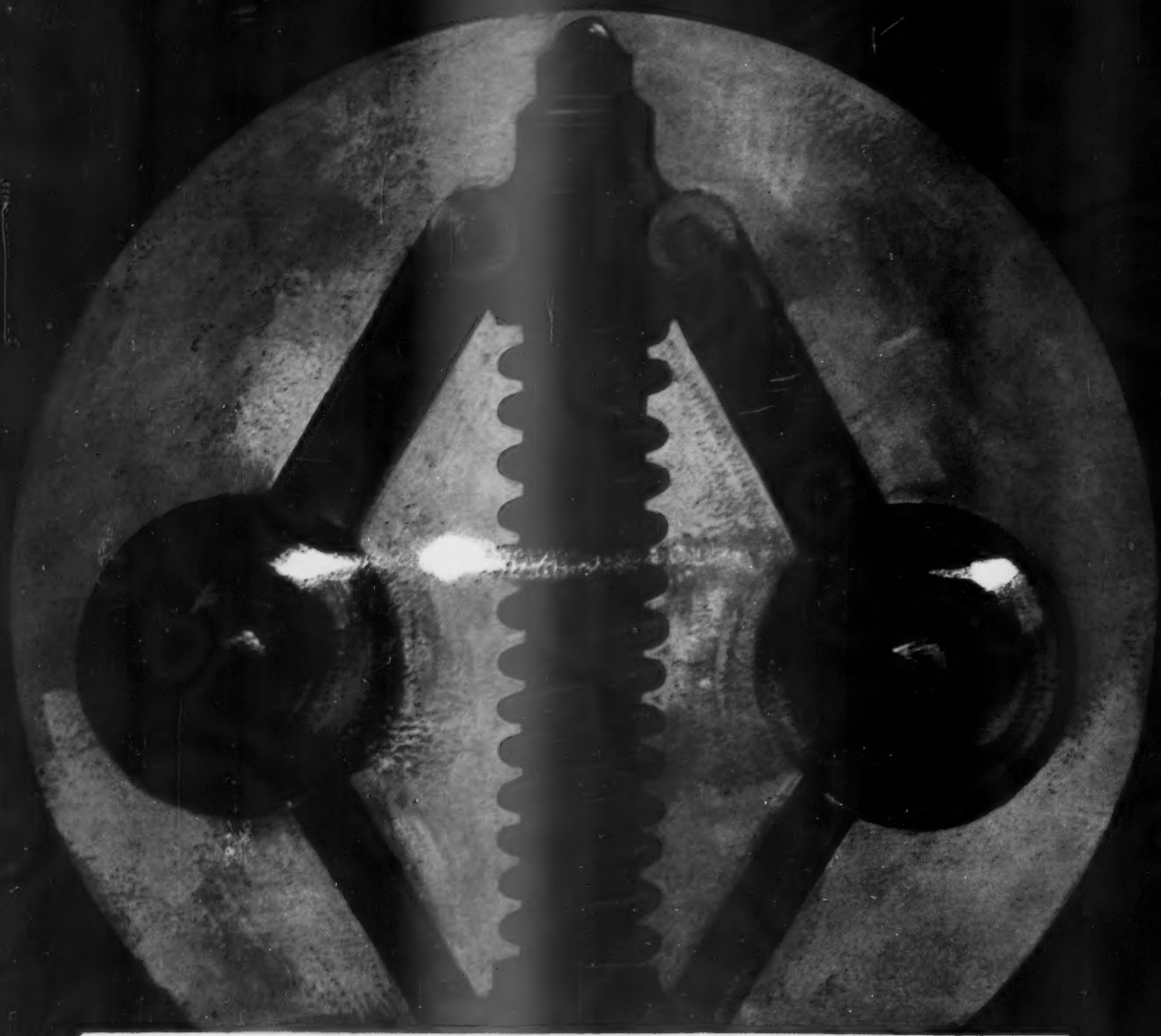
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The static friction which hampers the sensitivity of so many types of governor is eliminated in the Iso-Speedic governor by the use of flybobs consisting of hardened steel balls running on ground steel tracks.

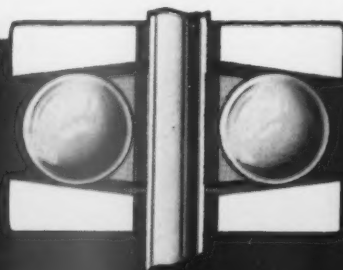
As a result, Iso-Speedic governors can be supplied which give control within 0.3 per cent and are used on generators for radar and television.

Other Iso-Speedic governors are available where a lower degree of accuracy is sufficient. Iso-Speedic governors are used on diesel engines and petrol engines and are available for other speed control applications.

The services of our engineers are at your disposal.

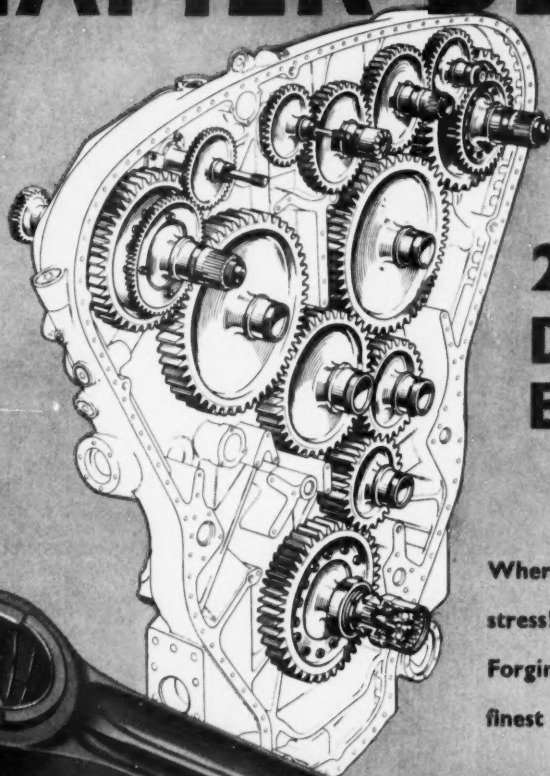
THE ISO-SPEEDIC COMPANY LIMITED, COVENTRY

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
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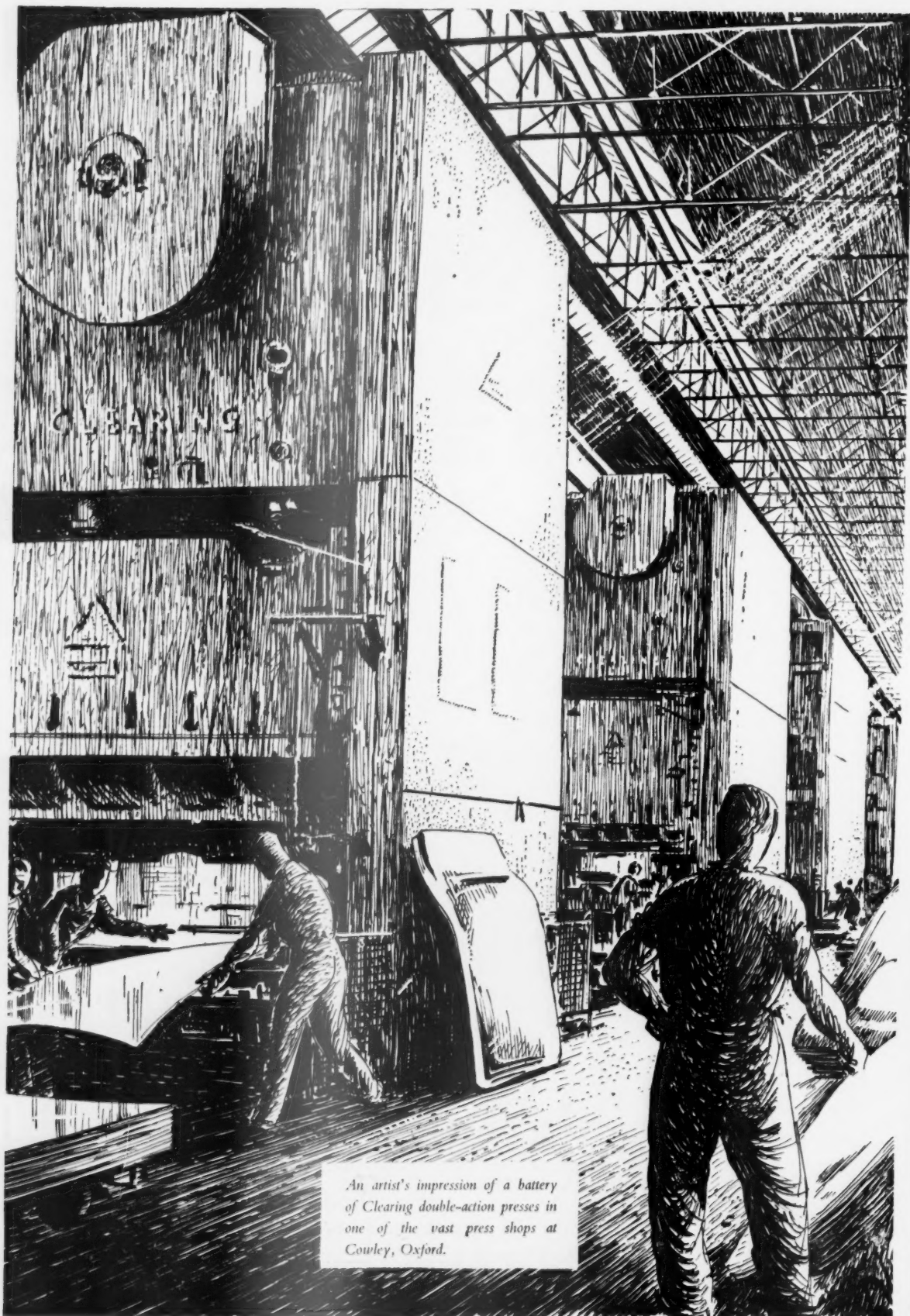
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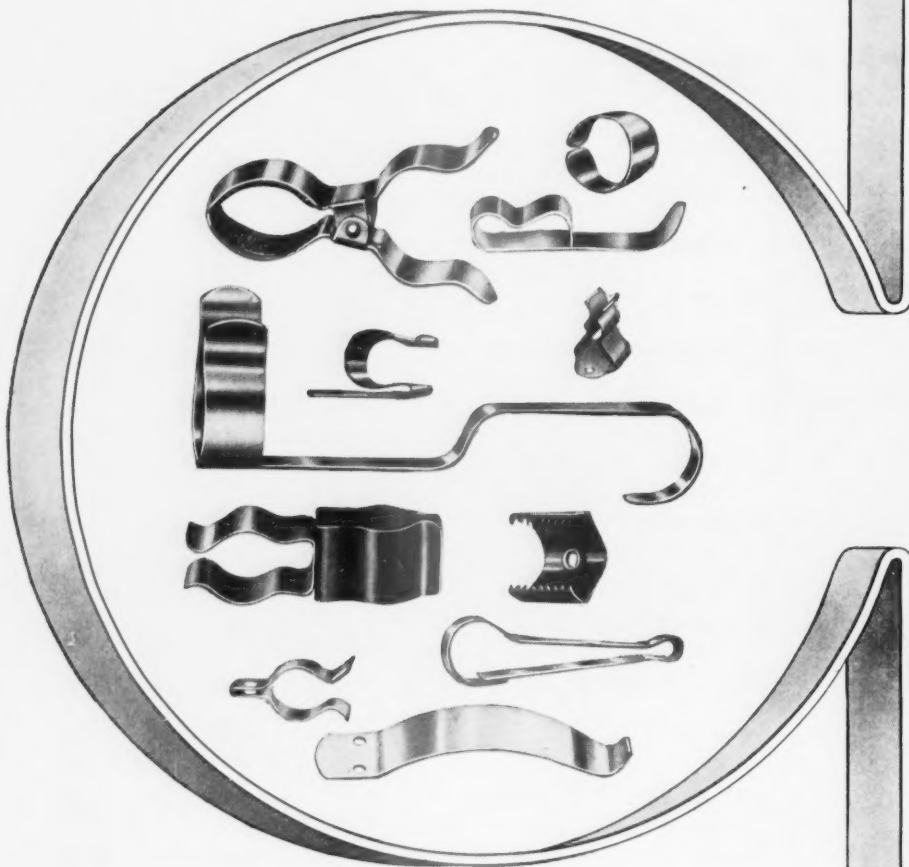
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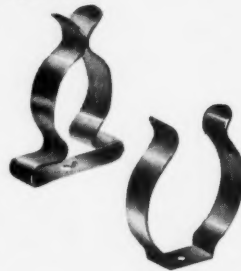
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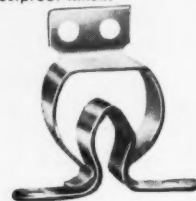
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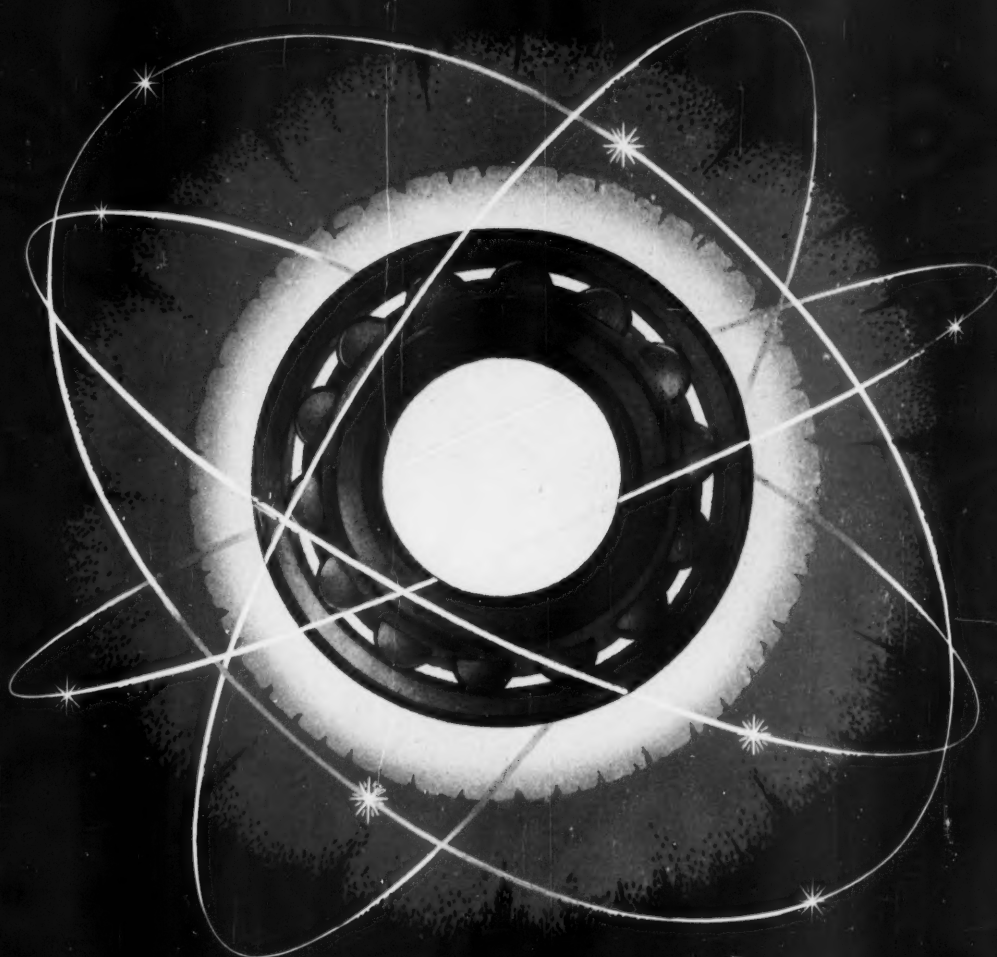
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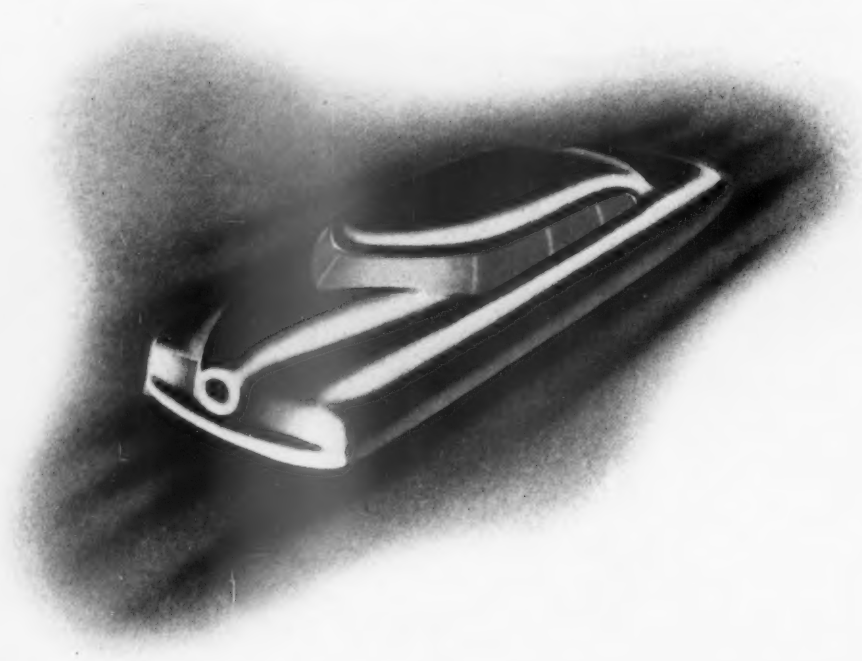
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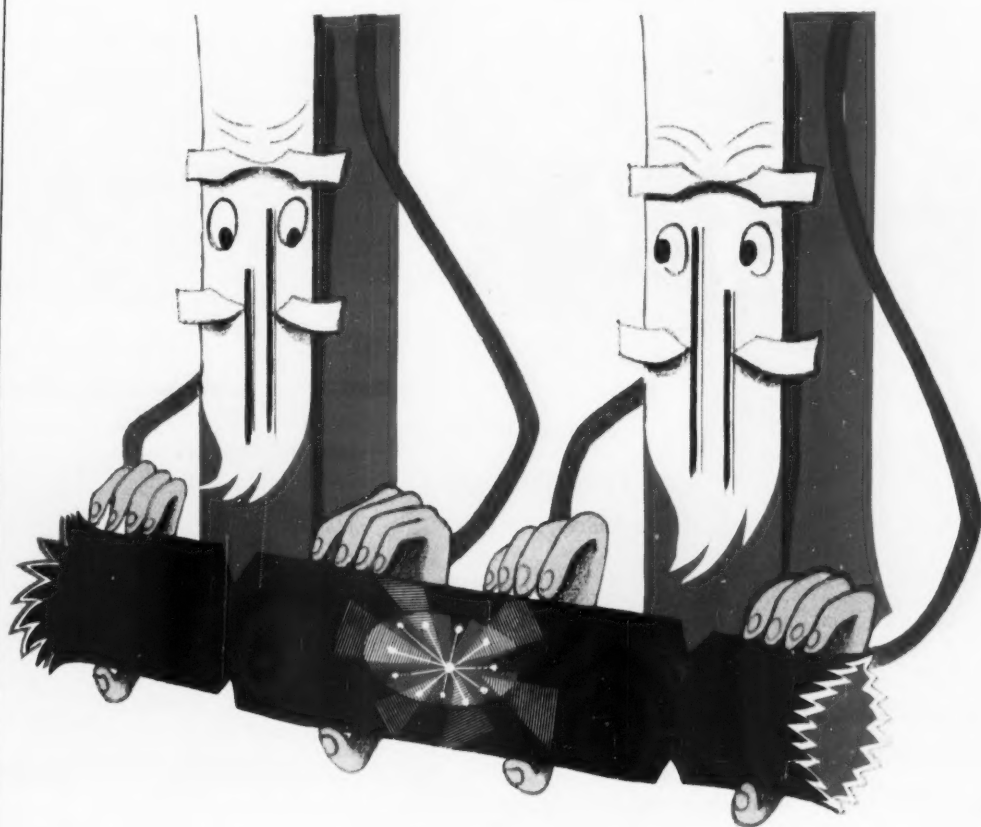


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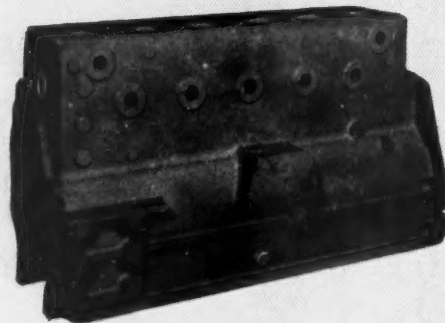
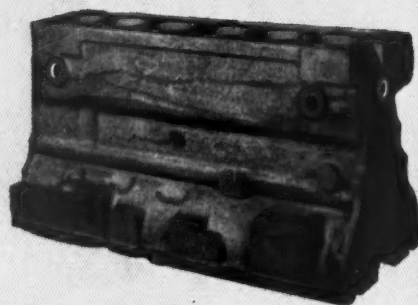
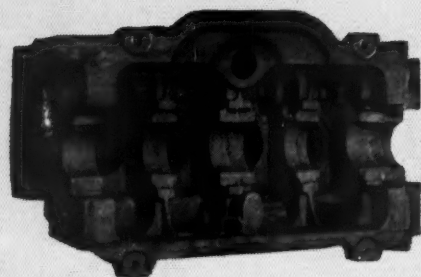
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Some gastronomic gauge, Comparator or Jet,
Which would discreetly take the measurement
Of all the food we — well but so unwisely — ate.*

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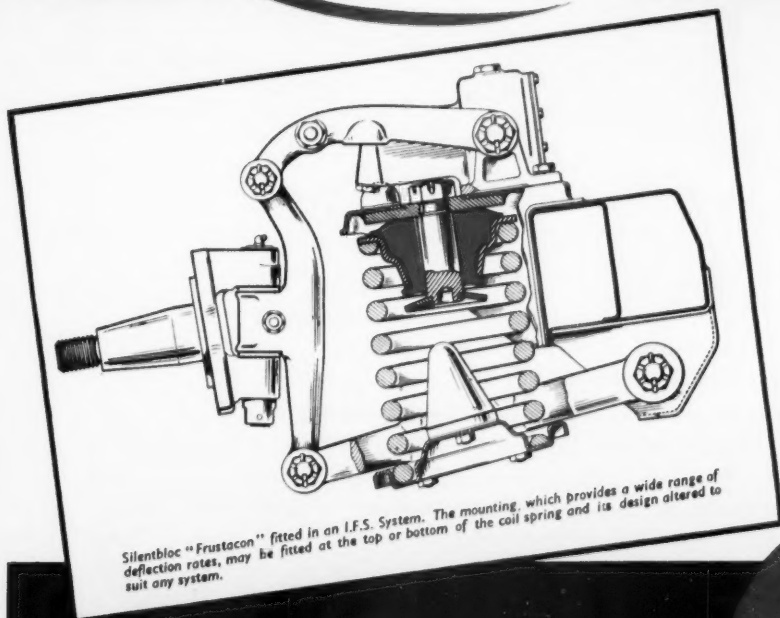
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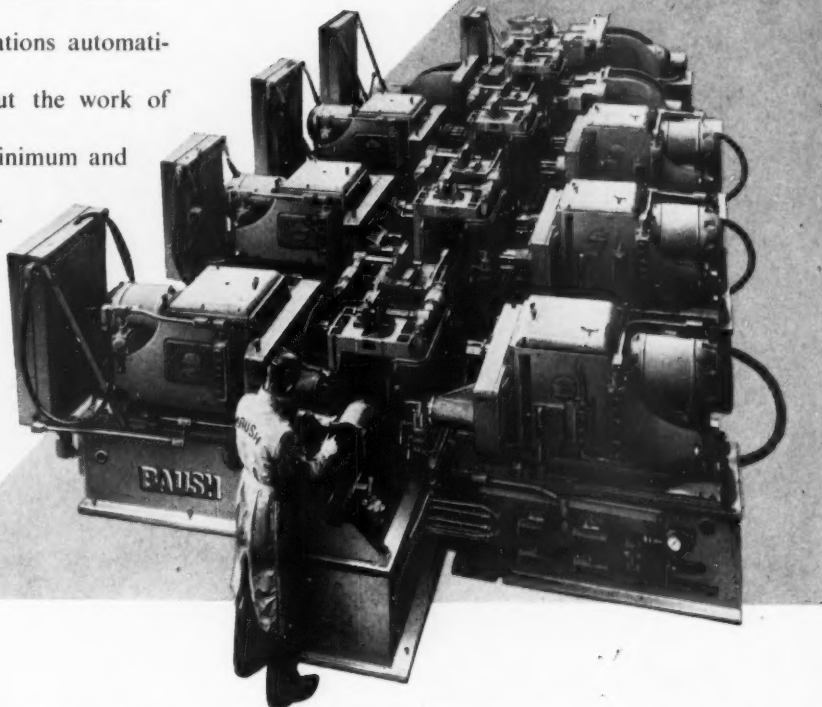
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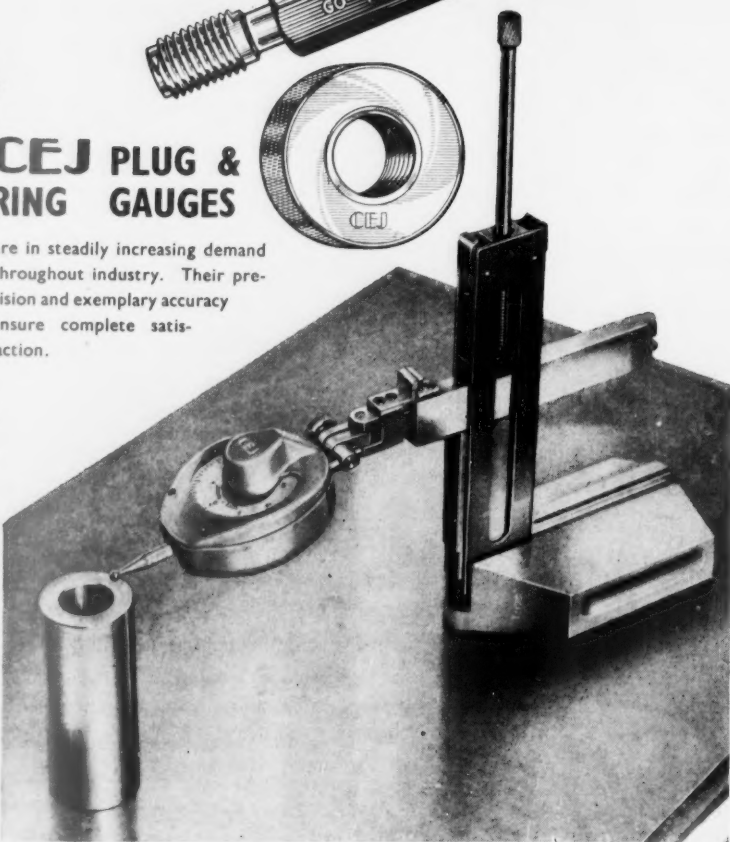


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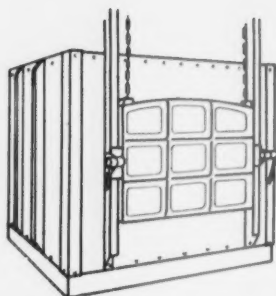
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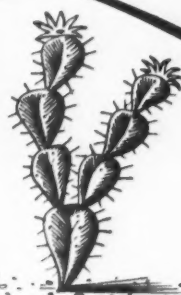
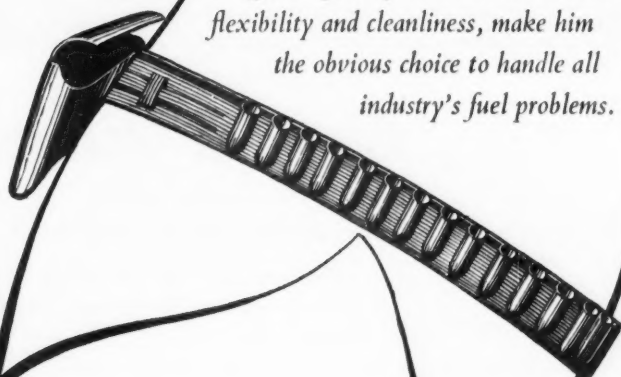
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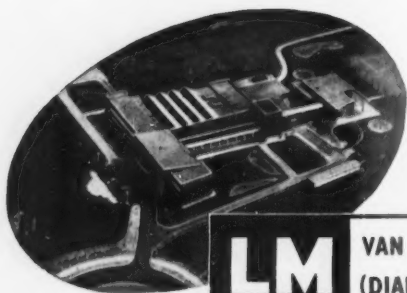
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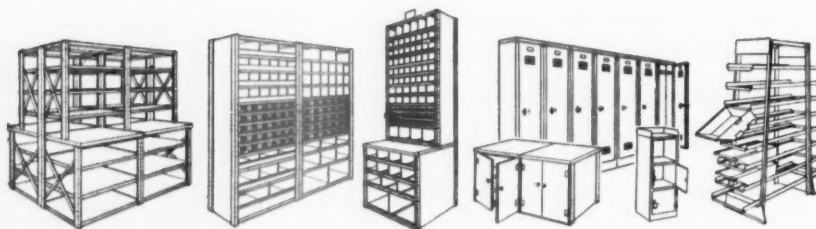
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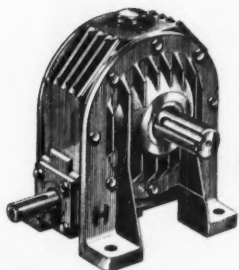


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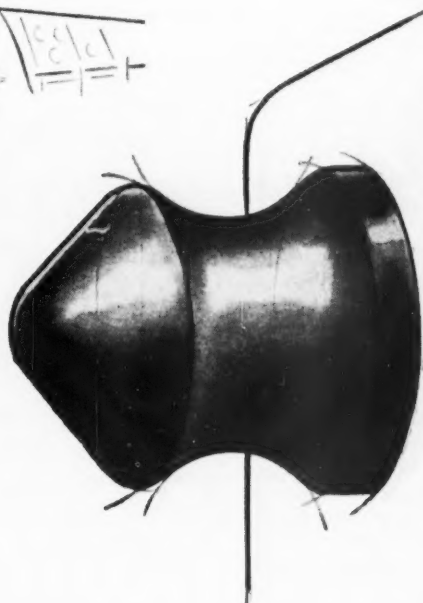
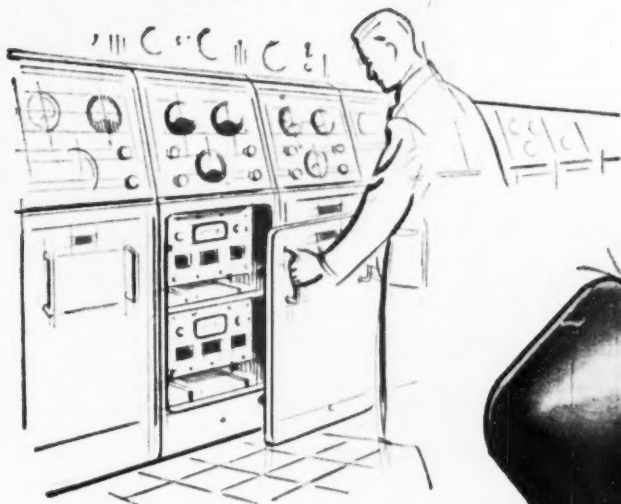
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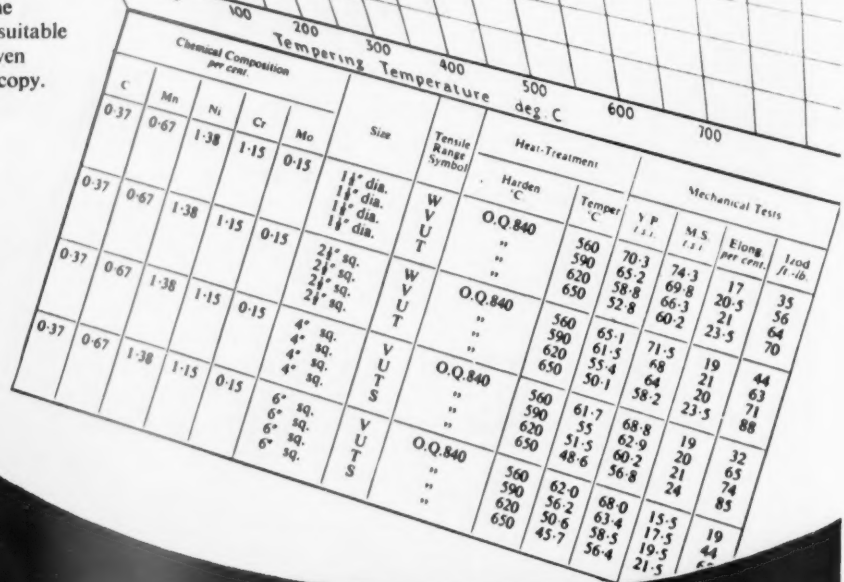


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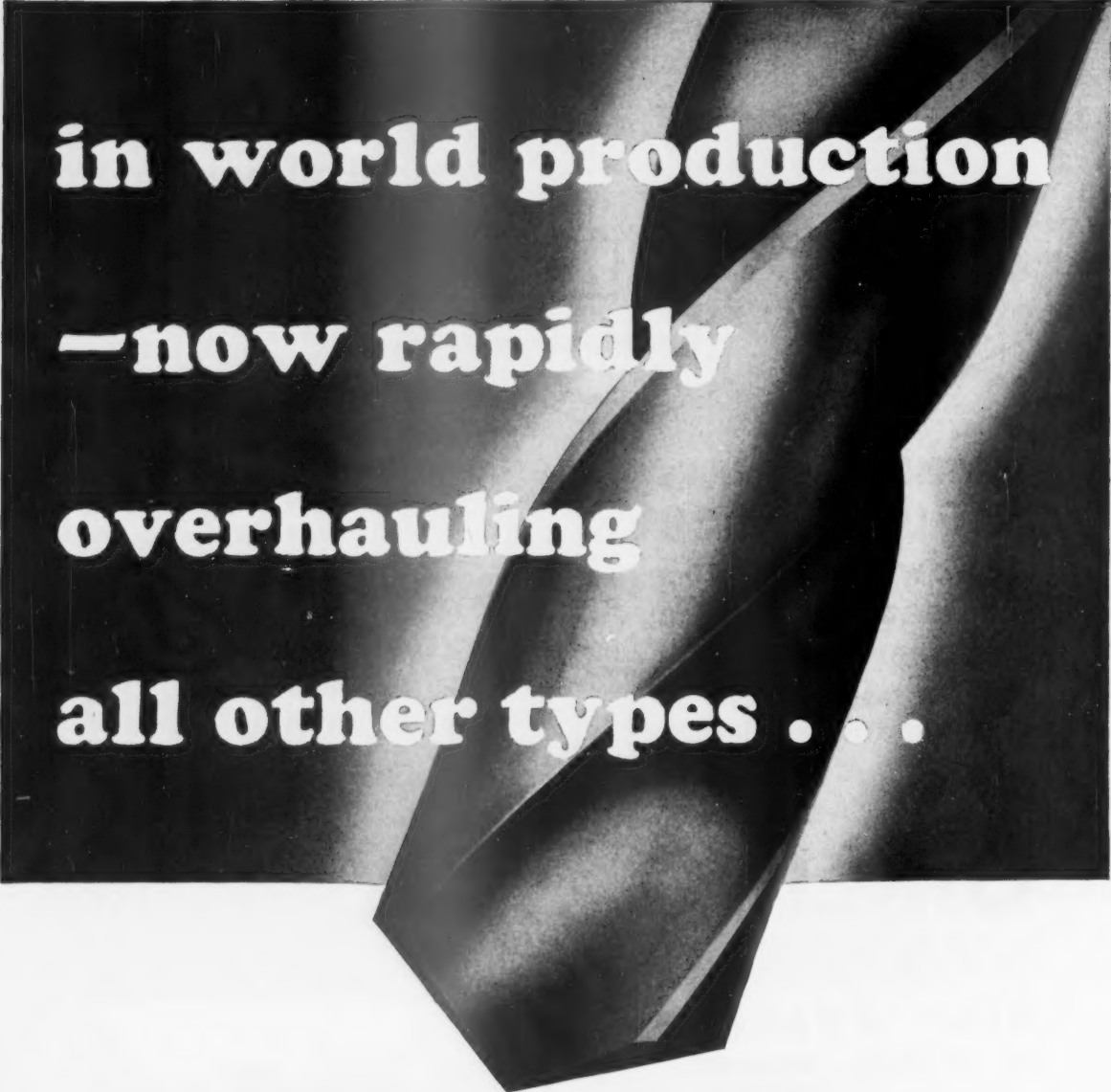
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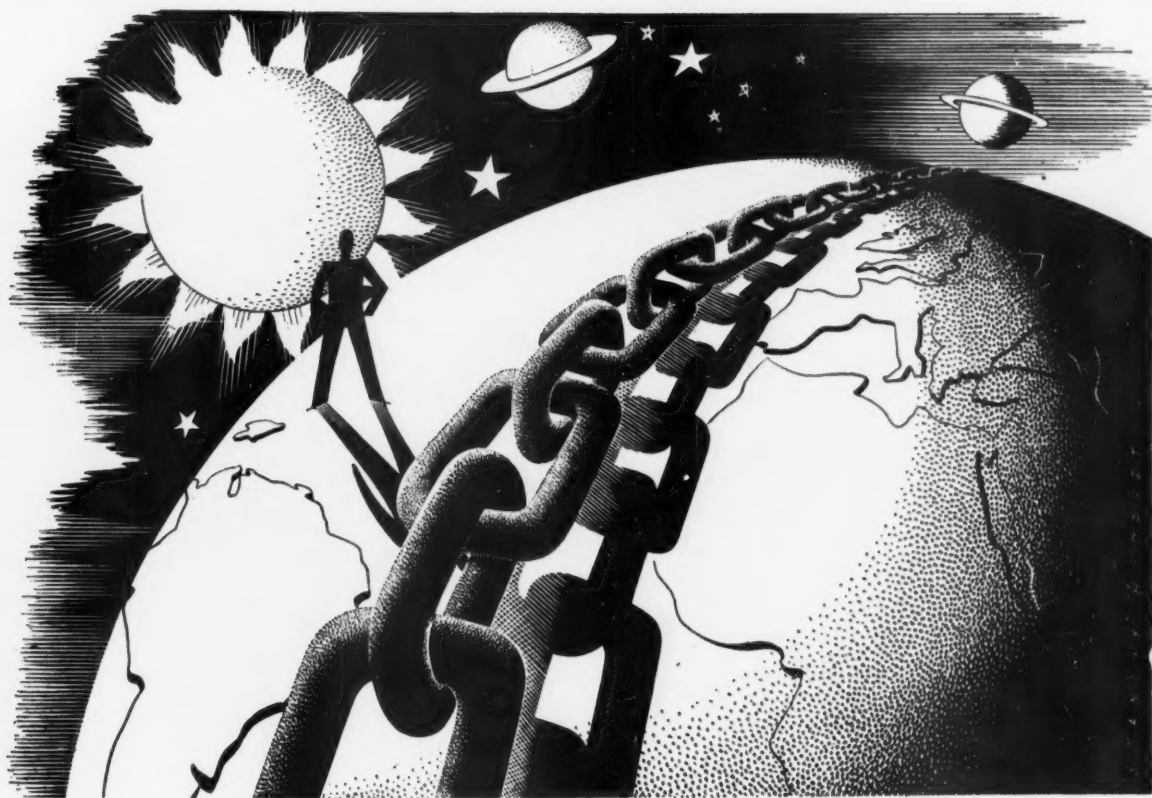


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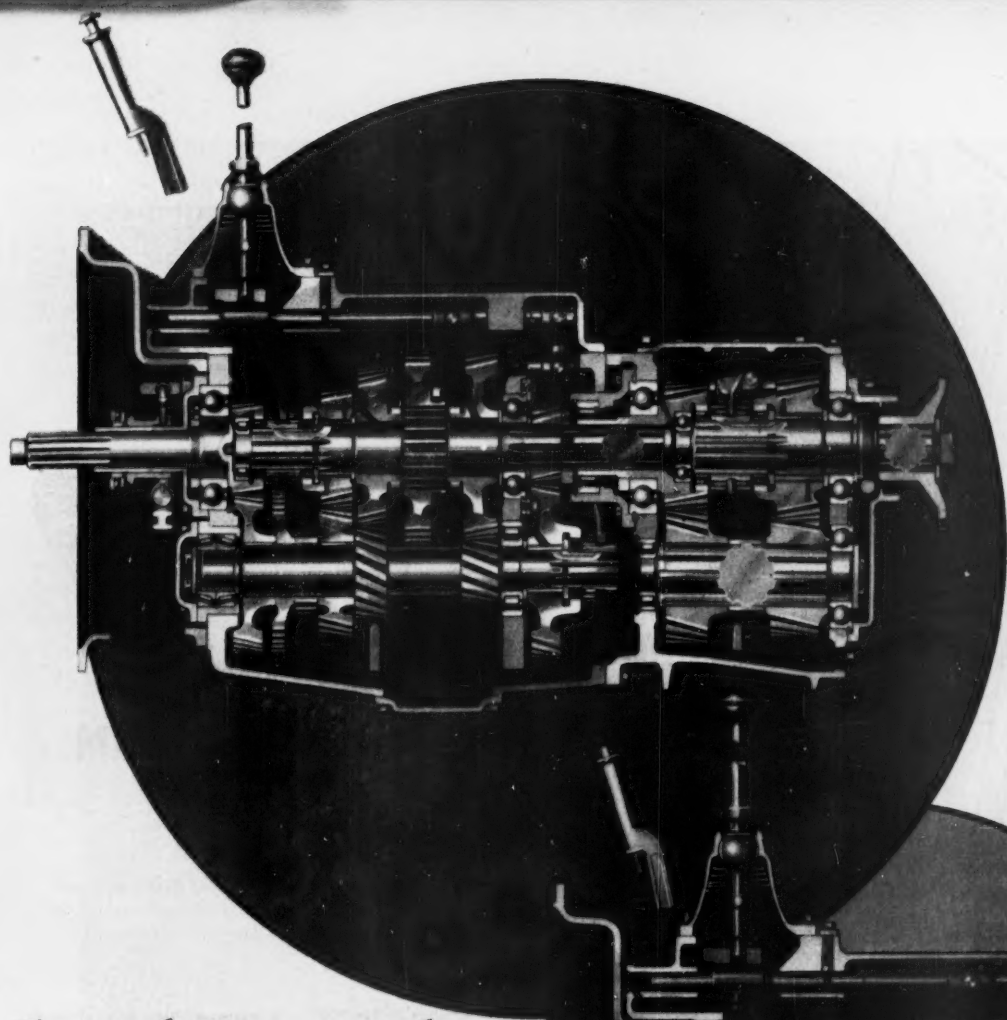
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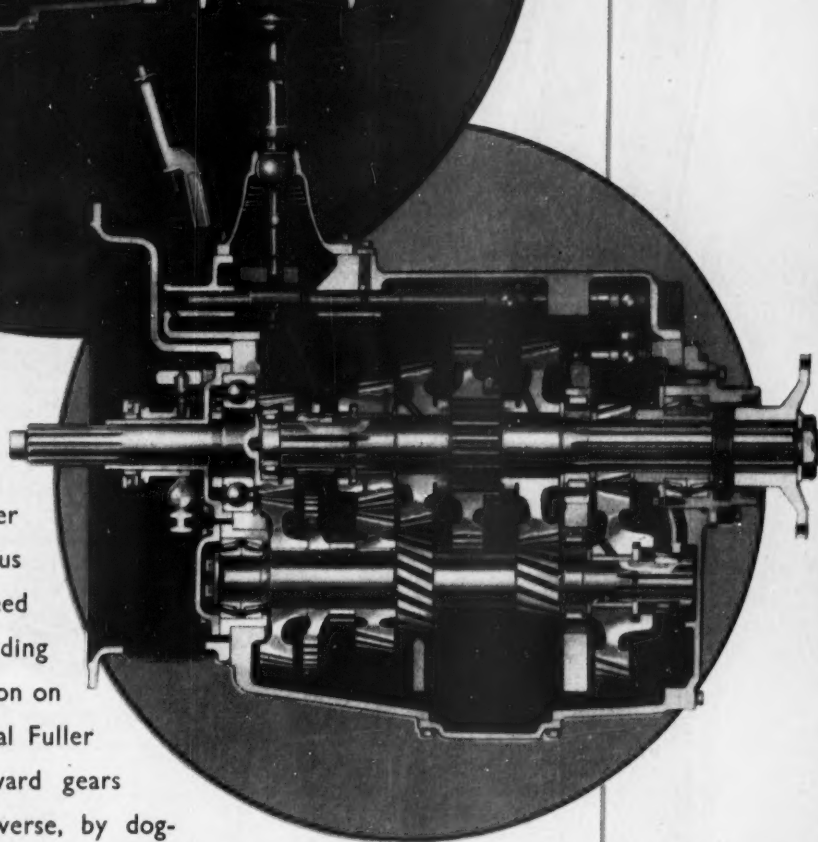
M.310



5 and 10-speed gear-boxes

The top illustration shows the Fuller 10-speed gear-box, comprising the famous five-speed box with a Fuller two-speed auxiliary box built on to it, thus providing a ten-speed box for heavy-duty operation on large trucks. This unit is to the usual Fuller standard of high-duty, with all forward gears helical, and all changes, including reverse, by dog-clutches. On both of these boxes the gears are shot-peened and crown-shaved, to avoid stress concentration.

The lower illustration shows the Fuller 5-speed gear-box. Every gear is helical and engaged by dog-clutches and to reduce shaft deflection to a minimum the mainshaft is supported on three bearings, and the layshaft kept short, making one of the highest-duty gear-boxes ever produced.



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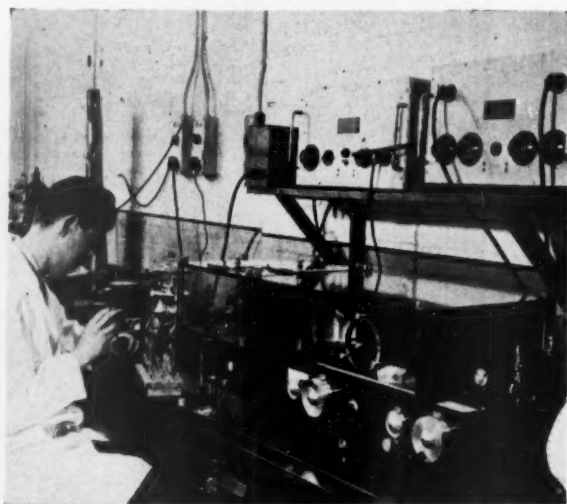
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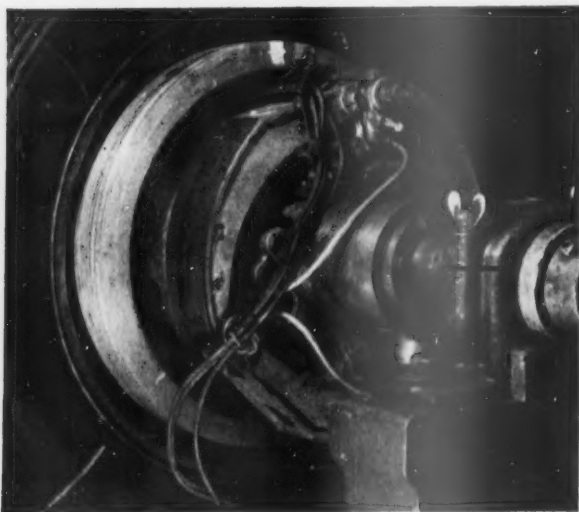
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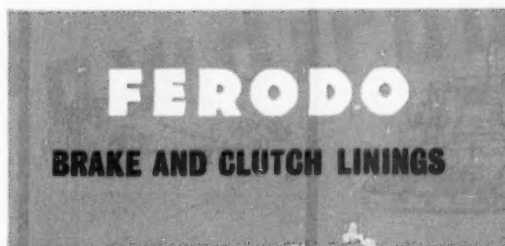


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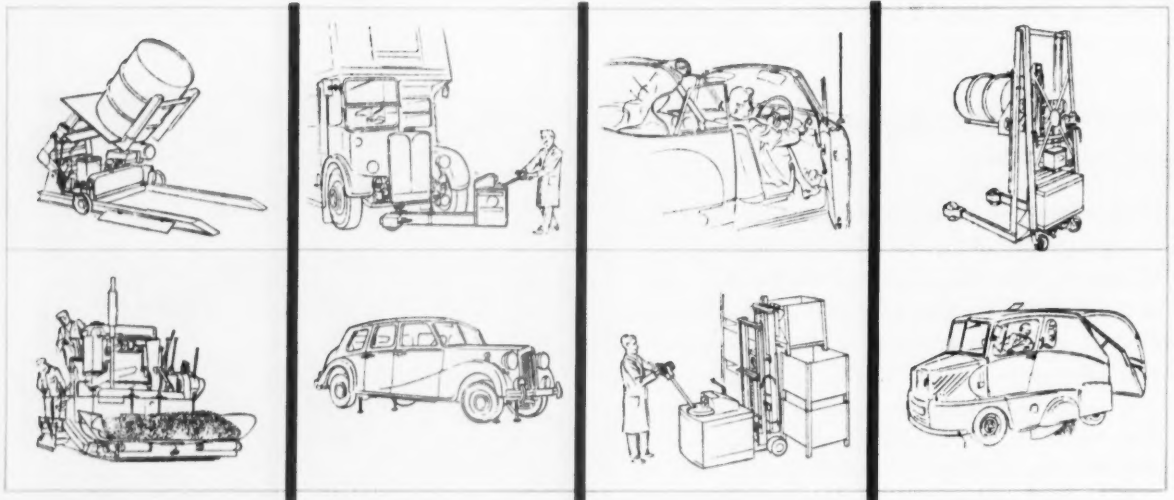


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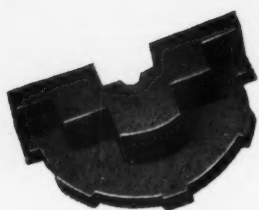


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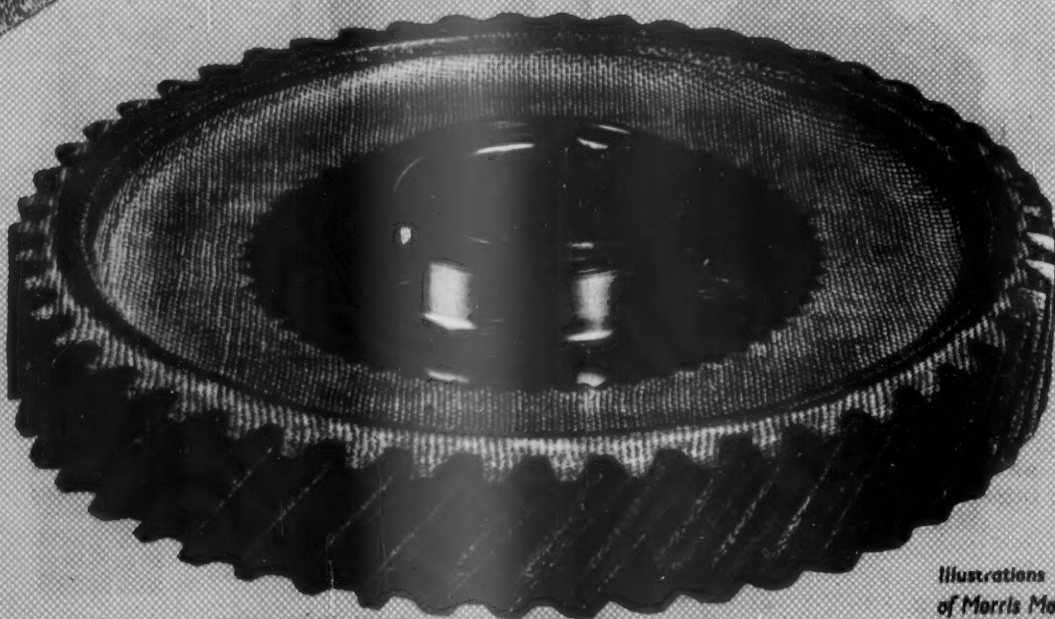
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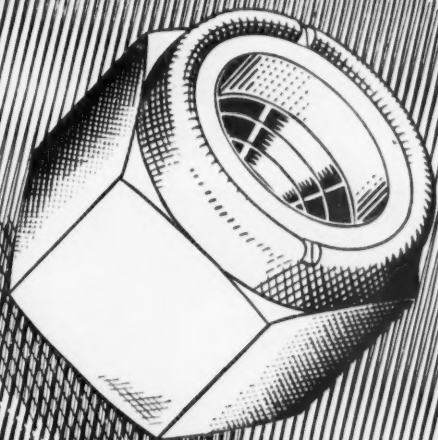
The illustrations show a sectioned gear blank and the finished spur gear



Illustrations by courtesy
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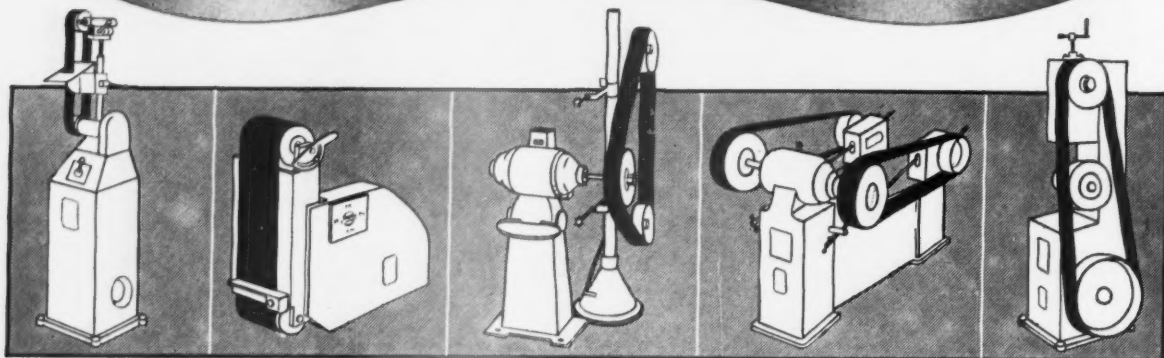
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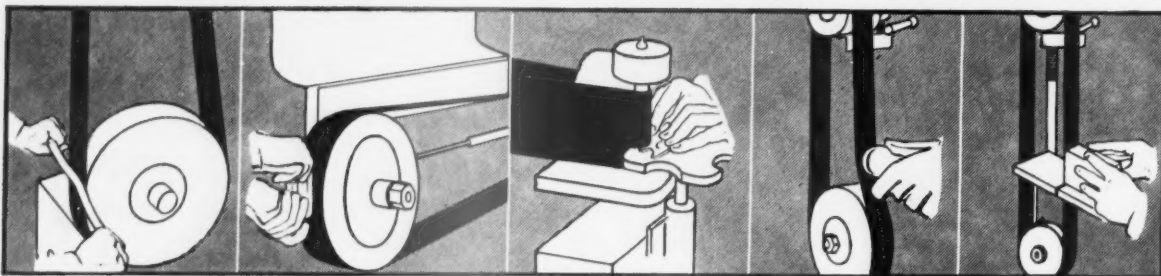
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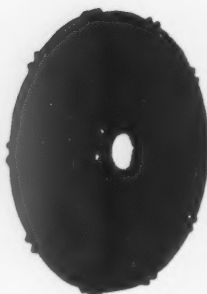
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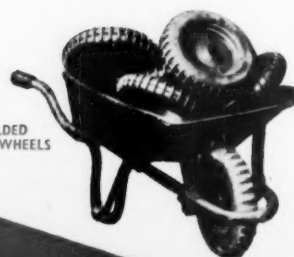
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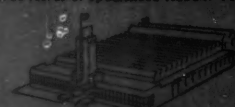


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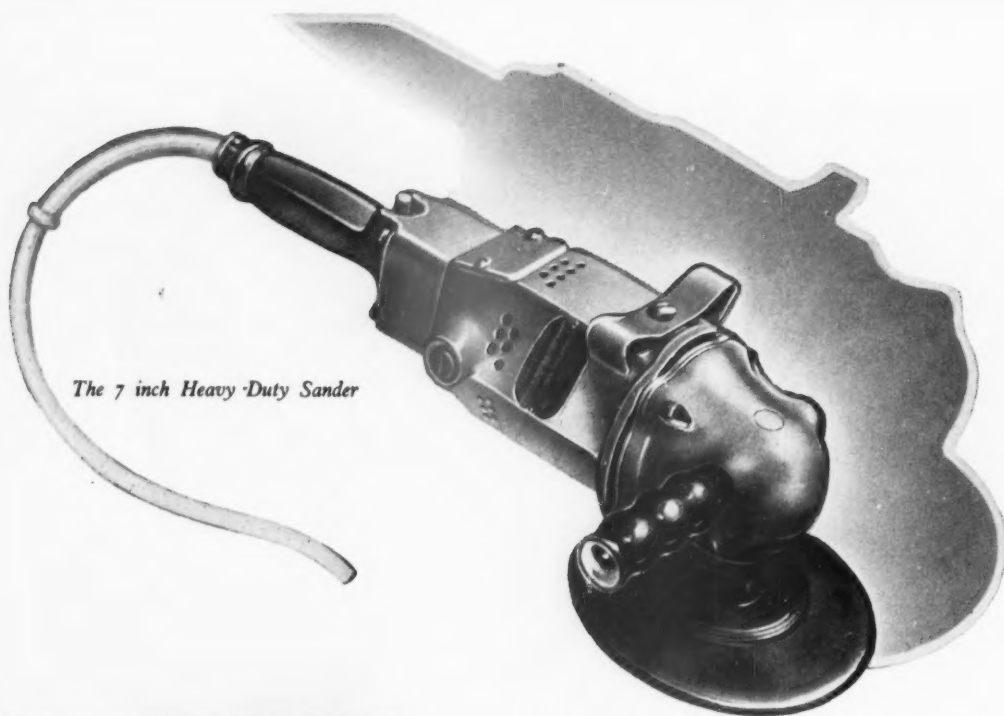
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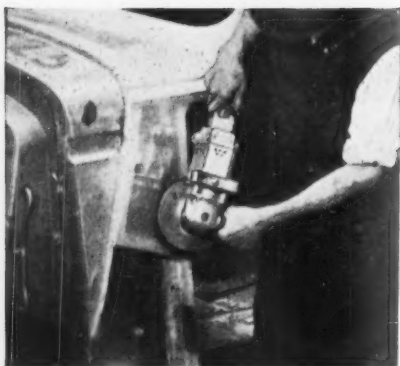
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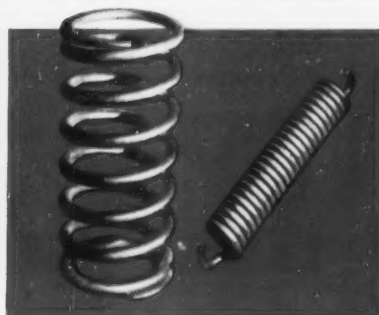
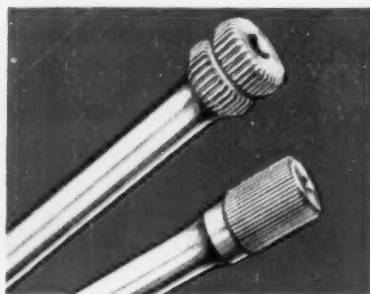
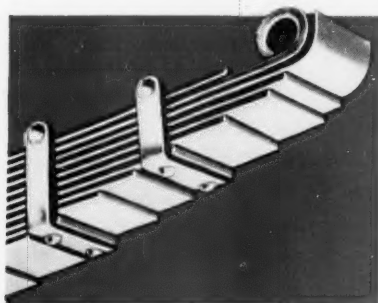
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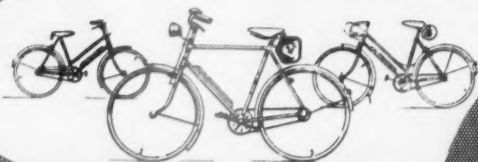
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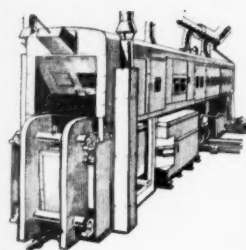
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NIMONIC 80
NIMONIC 80A
NIMONIC 90
NIMONIC 95
NIMONIC DS

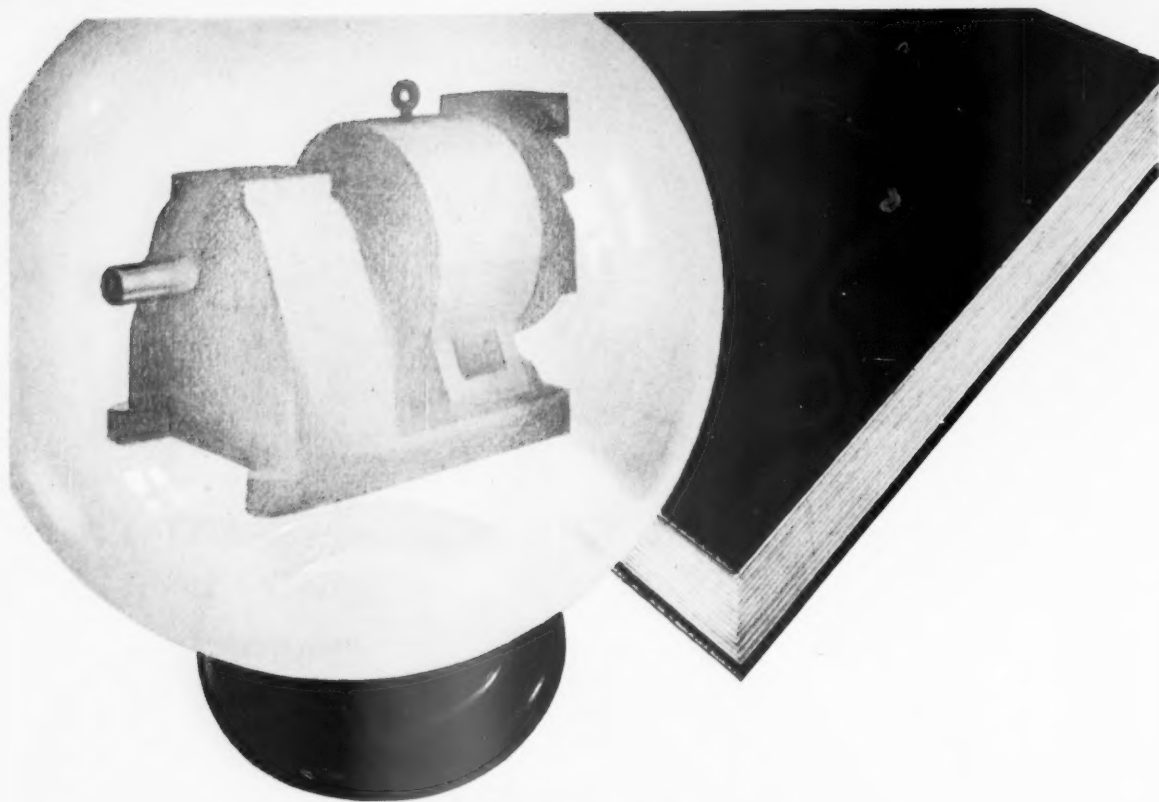
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G.42

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Design, Materials, Production Methods, and Works Equipment

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Chassis Lubrication

MANUFACTURERS tend to be so engrossed with their immediate design and production problems that considerations relating to servicing operations to be performed by someone else at a later date are often not treated as methodically as they ought to be. This is particularly true of chassis lubrication. The situation at present is that not only are there on many models too many lubrication points, but also the recommended periods between lubrication operations vary enormously as between the products of different manufacturers. Moreover, a wide variety of lubricants sometimes is specified for use on each chassis.

At first sight, it would appear that standardization of recommendations to be made throughout the industry might be a fairly simple matter. However, this certainly is not so; nevertheless, much might be gained by co-operative study of the problem by the servicing trade, commercial vehicle operators, motor manufacturers and oil companies. Although, in this country, commercial vehicle operators and the trade generally would derive most benefit from any positive steps taken as a result of such a study, manufacturers would also find them to their advantage; for not only would simplification of lubrication requirements in itself gain for them much goodwill from their customers, but also servicing attention would tend to be better carried out and wear and tear would be reduced, so that further gains in goodwill would be made. Moreover, in some overseas markets, notably in the Americas, motor cars are expected to function for long periods with an absolute minimum of attention. Therefore, improvement in this direction would be a good selling point.

When laying down rules for servicing organizations, it is necessary, as in other matters, to consider carefully the human factor. At present the garage mechanic finds, for instance, that grease is specified for the lubrication of a certain component on one make of car, whereas for an identical part made by the same component manufacturer, but fitted in a different make of car, oil is specified. It is only natural, therefore, that he should be misled into believing that almost any lubricant will be satisfactory. As a result, he tends to use any grease or oil that he thinks fit and thereby limits his lubricant stock to a small variety. In any case, if he were to keep all of the wide range of lubricants recommended for the vehicles to be serviced, he might have no room left for other essential stores. The result of this is that servicing is often ineffective, and rapid

wear and failures result, to the detriment of the good name of the motor manufacturer.

The technical problem

Technically, it is impossible to specify one, two or even three oils or greases to cover the requirements of all lubrication points. Nevertheless, it might be possible to do more in the matter of classifying lubricants into grades recommended for specific applications. One standard list agreed upon by all concerned would tend both to change the present condition of chaos to one of order, and to raise the general standard of servicing.

Many difficulties arise when trying to restrict the varieties of lubricant to be specified for one chassis. For instance, whereas some manufacturers are satisfied that all the year round in this country only one grade of oil is needed for use in engine sumps, others specify a Winter and a Summer grade. Again, two grades are sometimes recommended for rear axles. Gearbox requirements fall between those of the engine and the axle, but frequently, to simplify maintenance, either sump oil is employed in the gearbox, or axle oil is specified. That there should be wide divergencies of opinion on this subject is not surprising because the properties essential for an engine oil are in some respects entirely opposed to those needed for a gearbox oil, and other objections apply to using axle lubricant in the gearbox.

Engine oil must retain a reasonable degree of viscosity at high temperatures; on the other hand, gear oil is required to operate at much lower temperatures and should not become too viscous when cold. Hypoid axles are now employed almost universally, and they require a special extreme-pressure lubricant. Unfortunately, some of the additives essential in oils for hypoid gears attack chemically the materials used for synchro-cones. Consequently, if a lubricant is to be specified as common to both the axle and the gearbox, some compromise is required. In fact, there seems to be at present a general trend towards the use of S.A.E. 80 oil for both axles and gearboxes. This is probably because lubricants of this viscosity are becoming more readily available; the S.A.E. 90 grade, hitherto widely employed in axles, tends to be too heavy for gearboxes.

Hand lubrication

So far as gun or hand lubrication is concerned, there are probably five fundamentally different types of application for lubricants, each with a different requirement. Rolling-

element type bearings call for a grease that will not channel and leave the balls or rollers dry. On the other hand, a high melting point grease is required in other bearings of the same type but which run at high speeds in components such as the wheels, where it is essential that grease shall not escape and get on the brake drums. Another application is to bush type bearings. These have to be lubricated by a grease or oil that will not easily squeeze out but which, when it does, will readily flow back between the surfaces at the unladen side, whence it may again be induced, by reversals of load or by rotation of the pin relative to the bush, to flow round to the other side. The fifth application is to items such as Bowden cables and other controls which cannot be sealed and, therefore, are liable to dry out. In these, graphite and grease is generally specified.

The problem of rationalization is further complicated by the fact that manufacturers of some of the components refuse to guarantee their products unless certain lubricants are employed. Nor can the marketing policies of the oil companies be ignored. Nevertheless, all, no doubt, could fall into line.

Decisions as to what periods between lubrication should be specified are always difficult because widely differing operating conditions must be catered for. There are two conflicting requirements: one is that the car owner or vehicle operator should not be asked to do more than is absolutely essential, and the other is that there shall be adequate safeguard against excessive wear arising from under-lubrication. Nevertheless, when, as is at present the case, one manufacturer specifies no lubrication attention for the first 1,000 miles, and another recommends greasing certain components including king pin bearings and steering rod joints every 250 miles, there is obviously something wrong. We believe that in this particular instance, attention definitely is needed every 250 miles, but the fact that this is so would appear to indicate that these points are overloaded and that better design would have made it possible to call for less frequent lubrication.

This indicates that it is desirable to design specifically to reduce the amount of servicing required. To attain this end, it is necessary not only to provide adequate bearing areas, but also to arrange the geometry of the various assemblies in such a way that unduly heavy loads are not applied to the bearings. Moreover, much can be, and in many cases has been done to reduce the number of lubrication points.

About seven years ago, one British manufacturer con-

centrated on the elimination of as many lubrication points as practicable. The measures taken included the incorporation of an automatic chassis-lubrication system, the use where possible, of rubber bushes, sealed bearings, and reservoirs to ensure that a supply of lubricant to certain bearings was constantly available. Even when all possible had been done, there still remained about fourteen points requiring attention. Moreover, it has subsequently been found that sealed bearings and other devices for retaining the lubricant are not always satisfactory and, in many cases, their use has been abandoned. In fact, there has been a tendency to revert to hand lubricated bearings, for instance, in the water pump, instead of fitting sealed ones which are supposed to require no attention.

Although sweeping simplification may not be possible, at least some degree of rationalization ought to be practicable. Apart from compiling a standard list of lubricants for each component or group of similar components, standardization of periods between lubrication operations is desirable. Real benefits, for manufacturers, the trade and the customers, would accrue from the adoption of such a scheme.

Heaters

EVEN in our own temperate climate, heaters in which engine coolant is employed in the heat exchanger are hardly satisfactory for some time following a start from cold. Moreover they are relatively expensive and difficult to install. A better source of heat supply is the exhaust. Unfortunately, with jacketed exhaust pipes, there is a grave danger of leakage into the car body. For this reason, arrangements of this type are illegal in some countries.

However, it seems likely that the danger could be reduced at least to negligible proportions, if not eliminated altogether. One way of doing this would be to cast integrally with the heater jacket a central tube, and then to press the unit on to a straight portion of exhaust pipe, preferably adjacent to the manifold. In this way, should the exhaust pipe burn or corrode through, the gas would tend to blow out between the outer periphery of the pipe and the inner surface of the central tube of the jacket, thereby giving warning of the failure. Moreover, it is unlikely that the central tube would corrode through before the relatively thin exhaust pipe deteriorates to such an extent as to become completely unusable.

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ARMSTRONG SIDDELEY SAPPHIRE ENGINE

A 3.4 Litre Unit with an Exceptionally High Power:Weight Ratio

THE engine for the Armstrong Siddeley Sapphire was, of course, designed specifically to conform with the general policy laid down for the chassis design. This policy was to aim at a high top speed and good acceleration, not with a view to making a sports car of the vehicle, but rather to permit high cruising speeds to be comfortably and economically maintained without overstressing the power unit. Since the car is capable of a relatively high speed, it follows that when cruising at 60-65 m.p.h. the engine speed is moderate, so the rate of wear of the cylinders, bearings, and other components is low, as also is the fuel consumption expressed in terms of pt/b.h.p.-hr.

To meet the design requirements specified above, a high power:weight ratio was essential and, since the car was to be marketed at a highly competitive price, this performance had to be obtained without resorting to expensive arrangements. The twin overhead camshaft layout, because of the light weight of the reciprocating components, as well as for other reasons, is undoubtedly the best for the very high speeds required of power units for sports cars. By restricting the maximum engine r.p.m. to a more reasonable figure, however, it is possible to obtain equally good torque characteristics with push rod and rocker operated valves. The push rod and low camshaft system was adopted because it is relatively inexpensive, a short and simple chain drive can be used and only one camshaft is needed.

During the development stage, a

SPECIFICATION

ENGINE: Six cylinders. Bore and stroke 90 mm by 90 mm. Swept volume 3,435 cm³. Firing order: 1, 5, 3, 6, 2, 4. Maximum b.h.p. 150 at 5,000 r.p.m. with twin carburetors and 125 b.h.p. at 4,700 r.p.m. with one carburettor. Maximum b.m.e.p. and torque respectively 140 lb/in² and 198 lb-ft at 2,000 r.p.m. with twin carburetors, and 131 lb/in² and 182 lb-ft at 2,000 r.p.m. with one carburettor. Compression ratio 7:1. Fully balanced, four-bearing, forged crankshaft. Overhead valves, push rod operated. Carburetors: twin DAA36 Stromberg units; or single DAV36 Stromberg unit, each with a choke diameter of 1 $\frac{1}{8}$ in. Hemispherical combustion chambers. A.C. mechanical fuel lift pump.

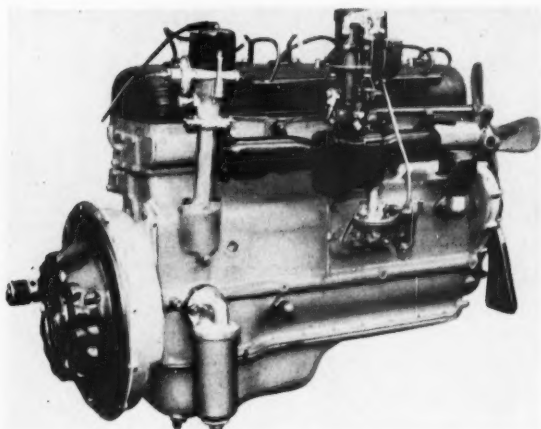
squish-type of combustion chamber, with in-line valves, was first specified mainly because it was thought that smoother running characteristics might be obtained as a result of the turbulence induced by this arrangement. However, it was decided also to try a hemispherical combustion chamber design, and this was found to give equally good results, so far as smoothness was concerned. Moreover, the greater power output obtainable with this layout was deemed fully to justify the extra cost arising from the necessity for employing two rocker shafts, one for the inlet and the other for the exhaust valves.

Although the weight of the unit had to be kept as low as possible, cast iron was specified for the main components

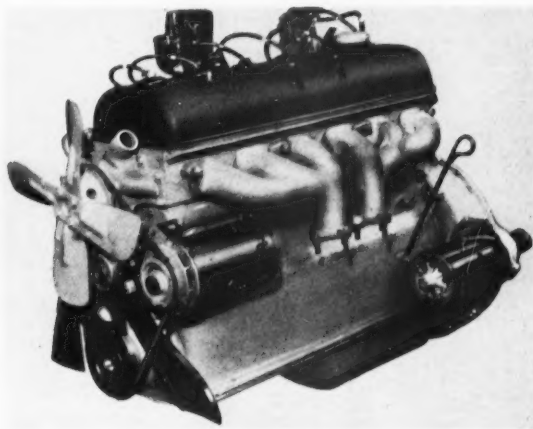
such as the crankcase and cylinder head. Had aluminium been employed, the cost would have been much higher. Another feature that has led to economy of weight is the square bore: stroke ratio. This has reduced the overall height of the engine, although it may have slightly increased the length. It has also led to a greater volumetric efficiency by making possible the incorporation of bigger valve ports. Because of the large diameter cylinder bores there is ample room for the main journal bearings.

The general arrangement of the main components is as follows: An integral cylinder block and crankcase casting carries the four-bearing crankshaft. On its right-hand side is the camshaft, which is positioned at a level approximately midway between the top and bottom of the cylinders. This height represents a compromise between conflicting requirements; on the one hand it had to be as high as possible in order that the push rods may be of minimum weight, while on the other hand, it could not be placed in too high a position otherwise the rods would have been inclined at too great an angle, relative to the rocker arms, for efficient operation.

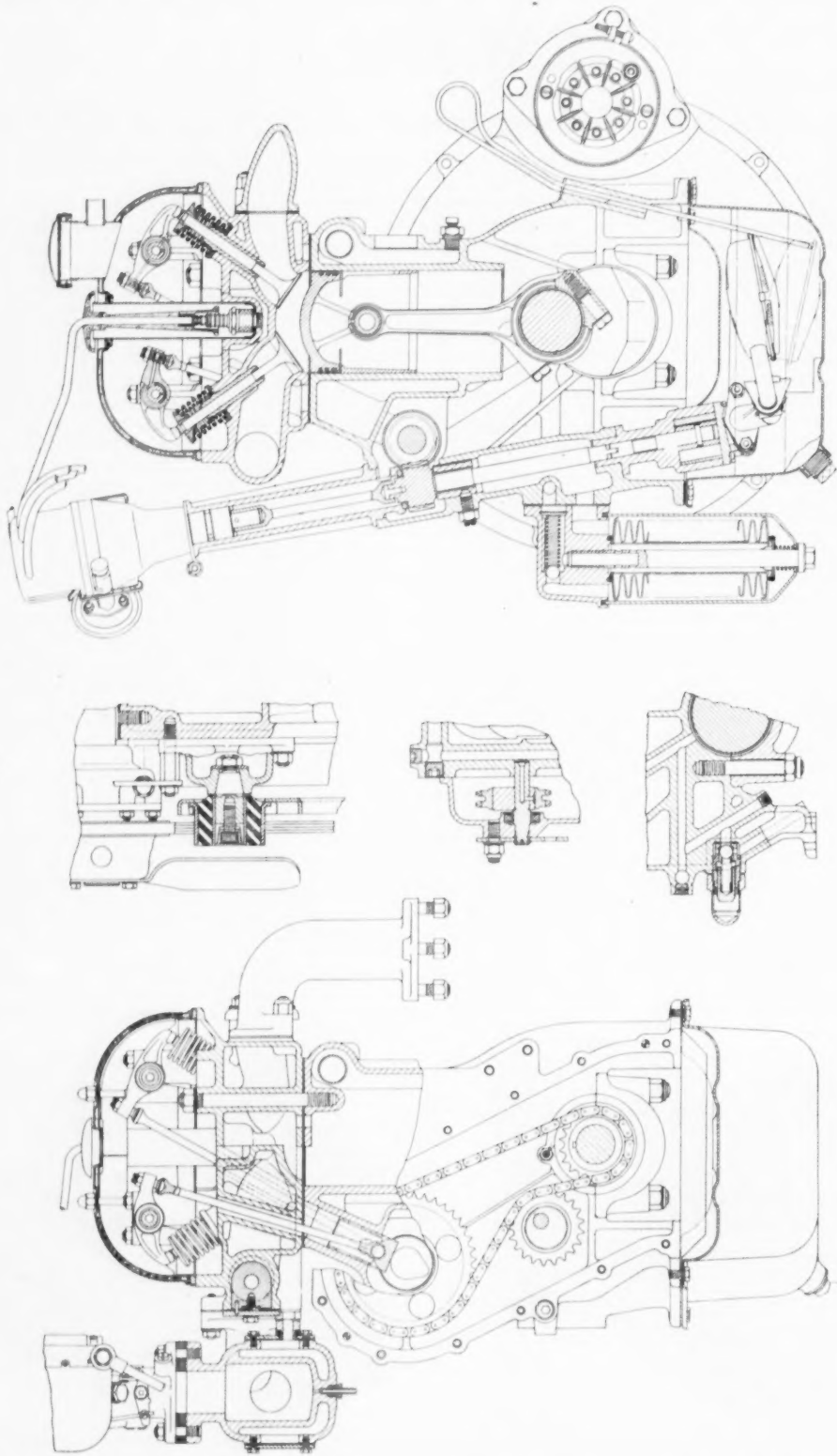
One large rocker cover encloses the valve gear. The two rocker shafts are carried on pedestals mounted on raised bosses on the cylinder head. These bosses are incorporated so that the face on which the pedestals seat and the joint face for the rocker cover may be machined in one operation. The induction manifolds are on the right and the exhaust manifolds are on the left.



The single carburettor installation on the right-hand side of the power unit for the Armstrong Siddeley Sapphire



A high-point type of mounting is carried on a bridge piece at the front of this engine



CROSS SECTIONS OF THE ARMSTRONG SIDDELEY SAPPHIRE ENGINE

The scrap views show the arrangements of the front mounting, jockey sprocket for the timing chain, and oil relief valve

A conventional triangulated V-belt arrangement is employed to drive the dynamo, and fan and water pump. The pump is bolted to the cylinder head, and the generator is pivot mounted on the left-hand side of the crankcase. Also on this side is the starter motor, which is bolted to a flange on the fly-wheel casing. On the right-hand side is the contact breaker and distributor unit, breather, oil filter and the mechanically operated fuel pump, which is actuated by an eccentric on the camshaft.

A bore:stroke ratio of 1:1 has been adopted, and the connecting rod length:stroke ratio is 1.834:1. When a single carburettor is employed, the mean piston speed at maximum b.h.p. is 2,776 ft/min at 4,700 r.p.m. The maximum b.m.e.p. is 131 lb/in² at 2,000 r.p.m., and the i.m.e.p. at the same speed is 152 lb/in². This gives a mechanical efficiency of 86.25 per cent. The power output is 2.11 b.h.p./in² piston area, and in terms of b.h.p./litre it is 36.4. Without the clutch and gearbox the engine weighs 574 lb dry, so the b.h.p. developed per pound is 0.218. This is an exceptionally good performance for an automobile engine. The minimum brake specific fuel consumption is 0.51 pt/b.h.p.-hr.

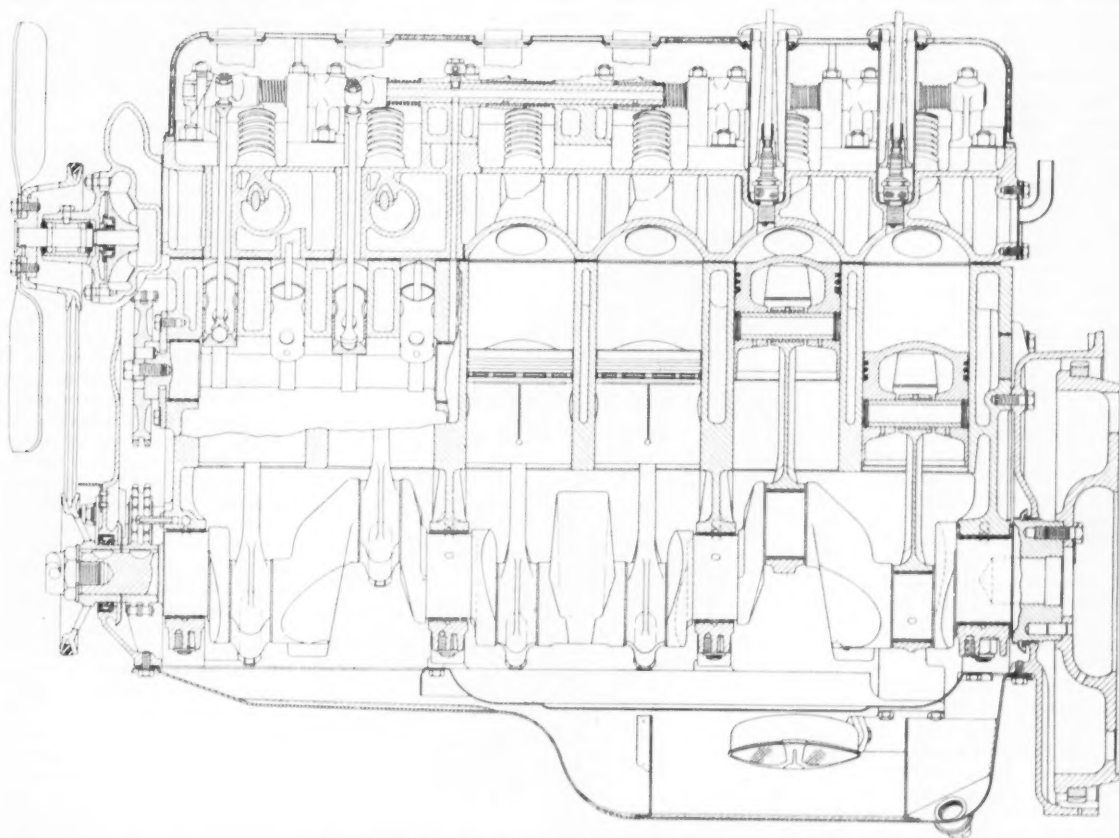
With two carburetors, the mean piston speed at maximum b.h.p. is 2,953 ft/min at 5,000 r.p.m. The maxi-

mum b.m.e.p. is 140 lb/in² at 2,000 r.p.m. while at the same speed, the i.m.e.p. is 160 lb/in². This gives a mechanical efficiency of 87.6 per cent. The power output is 2.54 b.h.p./in² piston area, and the b.h.p./litre is 43.6. Although the additional carburettor will add to the engine weight, the power output per pound should be appreciably better than that quoted for the single carburettor engine. The minimum brake specific fuel consumption is 0.515 pt/b.h.p.-hr. The overall height of the unit, less air filter, is 30½ in, its width is 22 in and the overall length, less flywheel, is 37 in. An engine installation angle of 1 deg 15 min relative to the horizontal plane of the frame has been adopted.

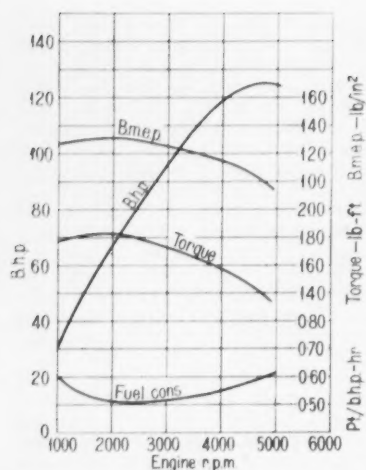
Metalastik engine-mounting units are employed. At the front, a single high-point layout has been adopted, a Metaxentric rubber-bush type unit being carried on a pressed steel bridge piece, approximately ¼ in thick, which is bolted to the frame side members. The axis of this bush is approximately coincident with the axis of oscillation of the power unit. The inner metal tube is carried on a trunnion pin, on one end of which is a taper and a ½ in diameter threaded extension for the nut which pulls it into a tapered hole in a small malleable iron casting bolted to the front wall of the cylinder block.

Accurate axial location of this front mounting is, of course, impracticable so, during assembly, the engine is lowered first on to its rear mountings and then the pressed steel bridge piece at the front is bolted to the frame. Then the inner tube of the Metaxentric bush unit is locked in position by means of an Allen socket-screw, in an axial hole in the trunnion pin, and a male and female split-cone arrangement. The socket-screw is countersunk into the inner cone to restrict the overall length of the mounting unit. This is necessary because it has to be accommodated in the space available between the fan and the cylinder block. It is the outer, or female, cone that is split, and the inner one is pulled into it until it expands and firmly grips the bore of the inner tube of the mounting unit. A small dowel in the split cone and the front end of the trunnion pin prevents relative movement taking place between them, which, during service, might cause the socket screw to work loose.

Although this single high-point type of front mounting is difficult to accommodate and calls for positive fore-and-aft location at the rear mountings, it has the advantage that it completely isolates the vehicle structure from the large amplitude rocking vibrations of the engine when running at tick-over speed. The more conventional V-type



Longitudinal section of the Armstrong Siddeley Sapphire engine.
Bore and stroke 90 mm × 90 mm. Swept volume 3,435 cm³



Performance curves for the engine when fitted with a single carburettor

layout does not do this entirely satisfactorily, because of the relatively large distance of the rubber units from the axis of oscillation. At the rear, the two Metacone units, one on each side of the gearbox rear extension, are bolted to brackets on a frame cross member. A pressed steel yoke over the top of the gearbox extension is bolted to the centre tubes of these cone mountings, which provide the necessary fore-and-aft location while allowing it to oscillate relatively freely.

Cylinder block and crankcase

The integral cylinder block and crankcase unit is of B.S. 1452 grade 14 cast iron. A finish of 25 μ in has been adopted for the cylinder bores, and the thickness of the walls is 0.23 in. The minimum space between the cylinders is 0.3 in, and the jackets extend up to the level of the top ring when the piston is at top dead centre. This is a good feature and should give adequate cooling at the upper ends of the bores where it is most needed. The skirt of the cylinder extends approximately 1 $\frac{1}{2}$ in below the jackets, but the cooling of this end presents no problems since at no time during the stroke is it exposed to the combustion gases. Cylinder wall distortion, arising from the loads applied by the cylinder head holding down studs, has been reduced to an absolute minimum by positioning the bosses for the studs symmetrically about the cylinder walls, and as far as possible away from them.

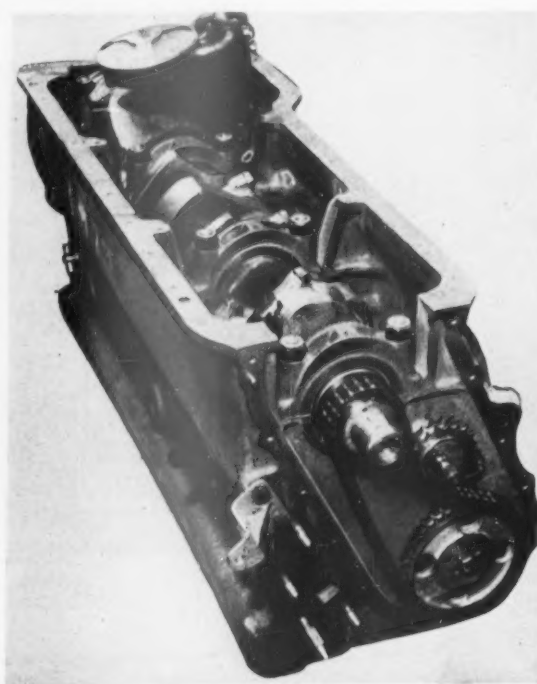
At the rear, a separate, cast aluminium alloy, flywheel housing is bolted-on and located by two $\frac{1}{2}$ in diameter silver steel dowels. No joint washer is fitted. A machined face near the bottom of this

casting forms part of the sump joint face, which is 3 in below the axis of the main journals, and also forms the oil seal behind the rear bearing cap. The front cover is also a bolted-on aluminium alloy casting, and its lower face is machined to form the joint face for the sump which seals in front of the adjacent bearing cap in a manner similar to that just described for the rear. An Oakenstrong joint washer is employed between the timing cover and the crankcase, while a Corok sump joint washer is employed.

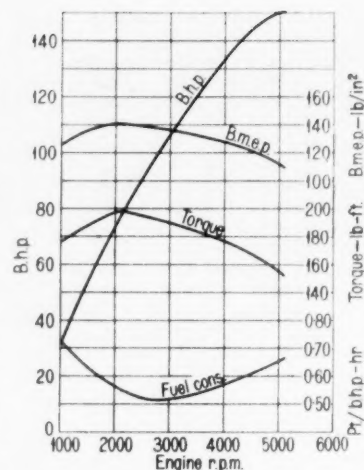
Two transverse webs and the end walls of the crankcase carry the four main journal bearings, as well as the four camshaft bearings the axes of which are at a level approximately $6\frac{1}{4}$ in above the axis of the crankshaft. The crankcase webs and end walls are stiffened by ribs extending radially from the bearing housings to a point on each side, at the junction between the crankcase side walls and the cylinder block. Two $\frac{1}{2}$ in diameter B.S.F. studs, of En 100T, and Simmonds Pinnacle self-locking nuts, hold down each bearing cap. These caps are laterally located between $\frac{1}{4}$ in high shoulders on each side of their seats on the housings. To facilitate their withdrawal for servicing, a hole for an extractor bolt is drilled and tapped in a boss below the centre of each cap.

Crankshaft, connecting rods and pistons

An En 9T crankshaft is employed, and it is balanced in the usual manner by weights cast integrally with the crank webs adjacent to the two end main journal bearings and by another



A deep-skirted crankcase is employed



Performance curves for the engine when fitted with two carburettors

balance weight midway between Nos. 3 and 4 crank pins. Semi-circular washers are fitted in grooves in front of and behind the rear journal housing to take the thrust. Each of the four main journals are 2.7505-2.7510 in diameter. Vandervell D2 bi-metal, babbitt-lined, shells are fitted and they are located in the conventional way by means of pressed-out tabs at the abutting faces of each half-shell. The effective bearing lengths are as follows: front and intermediate 1.245-1.255 in, rear 1.620-1.630 in. In all the main journal bearings the diametral clearance is 0.001-0.0025 in.

From the front of the front web to the back of the rear web, the length of the crankshaft is 24.9 in; the web thickness is 0.65 in, while the thickness of the inclined crank arms is 1 $\frac{1}{4}$ in. The diameter of the crank pins is 2.1250-2.1255 in. An oil-return scroll is cut round the rear of the shaft and works in a machined bore in a boss in the flywheel housing. The outer periphery of the inner end of this boss is lipped, and shrouded by a dish thrower ring pressed on a collar round the shaft. When a synchromesh gearbox is fitted, a Hoffmann 120 FS bearing is carried in the tail end of the shaft and supports the primary shaft. On the other hand, when the pre-selector gearbox is fitted, the end of the primary shaft is not supported in the tail end of the crankshaft, but is overhung from a Hoffmann LS11 bearing in a housing bolted to the front of the gearbox. This layout has been adopted because it has been found to lead to smoother operation of the centrifugal clutch than is possible with the more conventional arrangement.

A relatively light, 18-ton

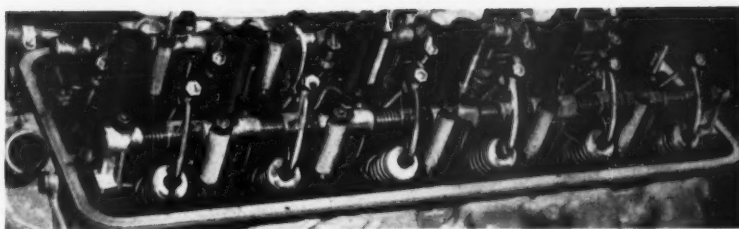
cast iron, flywheel is spigoted and bolted on the tail end of the crankshaft, and it is located by two $\frac{1}{4}$ in diameter silver-steel dowels. The overall diameter of the flywheel used with the synchromesh box is 14.336 in, the rim is 1.5 in wide by 1.56 in, and the unit weighs 41 lb. When the pre-selector gearbox is employed, the flywheel is of En 8 and it weighs 28 $\frac{1}{2}$ lb. Its diameter is the same but the rim is 2.2 in wide by 0.44 in. In both cases an En 8A starter ring gear with 142 teeth is shrunk on. This gear is heat-treated to a Brinell hardness figure of 255-277.

I-section connecting rods of En 8Q, with a centre-to-centre length of 6.499-6.501 in, are employed. The I-section tapers between the big- and small-ends from 0.90-0.80 in deep, the web thickness is 0.090-0.120 and the width over the flanges is 0.51-0.55 in. The weight distribution of the rod, complete with bearings, set bolts and tab washers, is estimated to be 1.4521 lb on the big end and 0.3789 lb on the small end.

The big ends are split at an angle of 40 deg from the axis of the rod and location is effected by dowel tubes round the $\frac{1}{4}$ in diameter, B.S.F., En 16U set bolts that secure the caps. These bolts are locked by tab washers. In the big ends, Vandervell D2 bi-metal, babbitt-lined shells are fitted. The length of the bearings is 1.120-1.130 in and the diametral clearance is 0.001-0.0025 in. A Vandervell Clevite B305 bush is pressed into the small end. It is 1.08 in long and the diametral clearance between the gudgeon pin and the bush is from 0-0.0004 in.

A high tensile steel, 2S14 gudgeon pin with a Rockwell hardness figure of C57-C63 is fitted. Its inside diameter is 0.625 in while the outside diameter is 0.8750-0.8752 in. In each of the piston bosses, the gudgeon pin bearing length is 0.89 in, and the diametral clearance, when cold is from +0.0001 in to -0.0003 in, that is, an interference fit. This ensures that, under sparsely lubricated conditions obtaining when starting from cold, relative motion takes place between the bush and the pin rather than between the pin and the piston boss. Axial location is effected by circlips in the bosses.

Brico slotted-skirt pistons of low-expansion aluminium alloy are employed. Each piston carries two compression rings and one oil control ring. The top compression ring is chromium plated while the second one is tapered and oxidized, and its top face is stamped with a T for identification. The dimensions of the compression rings are as follows: free gap, 0.49 in;



Inclined push rods actuate the rockers, which are carried on two shafts

when in the bore, the gap is 0.010-0.015 in; face width 0.0928-0.0938; radial thickness 0.136-0.143 in; depth of groove in piston 0.153-0.157 in; side clearance 0.0015-0.0035 in. The dimensions of the Maxigroove, oxidized oil control ring are the same as those of the compression rings, except in that the face width is 0.1553-0.1563 in. The complete assembly including rings, gudgeon pin and circlip weighs 1.50 lb.

starter handle. A Super oil seal is carried in the timing cover and bears on the boss of the pulley. The outer periphery of the rear end of the housing for this seal is lipped and partly shrouded by a pressed steel thrower ring which is clamped between the bosses of the pulley and timing drive sprocket. With this arrangement, oil can neither be splashed directly on to the seal nor can it run down from the outer periphery of its housing on to the seal.

A $\frac{1}{4}$ in pitch, two strand Renolds chain transmits the drive to the B.S. 1452 grade 14, cast iron half speed wheel, which is secured by a single En 16U, $\frac{1}{4}$ in diameter, B.S.F. bolt screwed axially into the end of the camshaft. This wheel is positively located by a $\frac{1}{4}$ in diameter dowel. Chain tension is maintained by an eccentric jockey sprocket. Adjustment may be made without removing the timing cover and it is effected in a somewhat unconventional manner. The eccentric is splined on to its spindle, the rear end of which is carried in a boss on the front wall of the crankcase, while its front end extends through a hole in the timing cover in which is housed a rubber seal round the spindle. A quadrant plate is splined on to the front end and retained by a circlip. Round the arc of this quadrant is a slot in which is registered a stud carried in a boss on the front of the timing cover. After the quadrant has been rotated to turn the eccentric and adjust the chain tension, a self-locking nut on the stud is screwed down on to it to fix the setting.

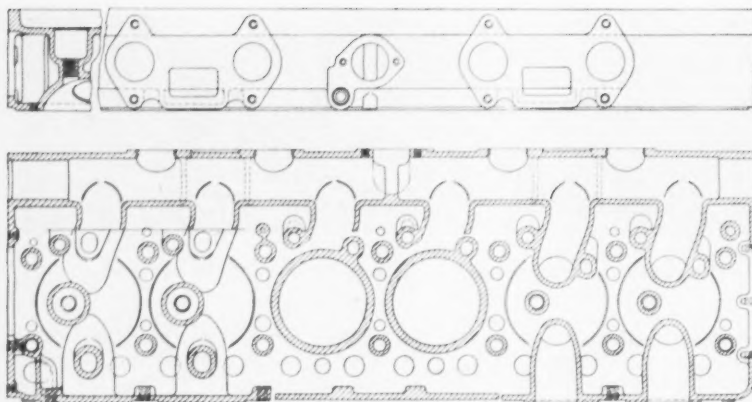
Four Vandervell, D2 bi-metal bushes are pressed into bosses in the crank webs and end walls to carry the En 32 camshaft. They are all 1.874-1.875 in internal diameter by 0.990-1.010 in long. The cams, journals, eccentric, and spiral gear driving the pump and distributor are carburized, hardened and tempered to a Rockwell hardness figure of C58-C62, and to such a depth as to give a minimum case thickness of 0.030 in after grinding.

VALVE DATA

	Inlet	Exhaust
Material	Silchrome	Silchrome XB
Head diameter	1.695-1.700 in	1.485-1.490 in
Throat diameter	1.45 in	1.345-1.350 in
Stem diameter	0.340-0.341 in	
Diametrical clearance	0.0005-0.002 in	0.0012-0.0027 in
Seat angle	45 deg	
Valve seat material	Cut in cylinder head	
Spring material	DTD 5A	
Spring rate, inner	46.75 lb/in	
outer	75 lb/in	
Spring length, free, inner	2.10 in	
outer	2.44 in	
Spring length, installed, inner	1.437 in	
outer	1.566 in	
Number of coils, effective, inner	7 $\frac{1}{2}$	
outer	5 $\frac{1}{2}$	
Coil diameter, mean, inner	0.782 in	
outer	1.144 in	
Wire gauge, inner	12 S.W.G.	
outer	9 S.W.G.	
Valve lift	0.3588 in	0.3075 in
Rocker ratio	1.28 : 1	1.10 : 1
Valve crash speed	5,200 r.p.m.	
Valve guide material	B.S. 1452 Grade 14	
Valve guide length	2.40 in	
Inside diameter	0.3415-0.3420 in	0.3422-0.3427 in
Outside diameter	0.6260-0.6265 in	0.6260-0.6265 in
Tappet clearance, hot,	0.006 in	
Tappet clearance, cold for timing only	0.016 in	0.014 in
Valve opens	8 deg B.T.D.C./46 deg B.B.D.C.	
Valve closes	62 deg A.B.D.C./18 deg A.T.D.C.	
Ignition timing	7 deg B.T.D.C.	

Timing gear, camshaft and valve gear

Two Woodruff keys in the 1.3742-1.3747 in diameter front extension of the crankshaft furnish the drive for the cast iron fan belt pulley and En 9T timing gear drive-sprocket, which are held on by an En 9T special bolt. This bolt is screwed into the $\frac{1}{4}$ in diameter B.S.F. tapped hole in the end of the crankshaft extension, and on its head are incorporated the dogs for the



The inlet rail is cored in the cylinder head

Axial location is effected by a two-piece thrust plate which is bolted to the front wall of the crankcase, and which engages in a $\frac{1}{4}$ in wide groove machined round the shaft.

The cam profiles have been designed to give the following characteristics at 4,700 r.p.m.: maximum positive acceleration of the tappet, on the flank of the cam, 9,354 ft/sec²; maximum negative acceleration, on the nose of the cam, 3,000 ft/sec²; maximum velocity 9.42 ft/sec. At the cam, the lift from the base circle to the nose is 0.285 in, and the nominal period is 121 deg without the silencing ramp. This ramp has a period of 34 deg both for opening and for closing.

Piston type tappets of chilled cast iron are employed. They are carried in a tappet chest cast integrally with the upper portion of the cylinder block. The bores in which the tappets are housed are 1.12375-1.12425 in diameter and the diametral clearance is 0.00045-0.00195 in. Since they are not perpendicular to the top face of the block, their upper ends are counterbored to facilitate the final machining operation.

In the base of each tappet is seated a $\frac{1}{4}$ in diameter hardened steel ball on which is carried the cupped lower end of the $\frac{1}{4}$ in diameter En 8 push rod. Both ends of each push rod are cyanide hardened to a minimum of 207 Brinell, and sand blasted to give an oil retaining surface. These rods are not vertical; those for the inlet valves are inclined at an angle of 17 deg from the vertical, whereas those for the exhaust valves are inclined at 23 deg from the vertical. The effective length of the inlet push rods is 8.15 in while that of the exhaust rods is 9.85 in. By employing the lighter rod to actuate the heavier valve, and *vice versa*, it has been possible to make one pair of springs common to both inlet and exhaust valves.

Vandervell Clevite bushes are pressed into the bores of the rockers. The polished end-pads of the rockers, which bear on the valve stems, are case hardened to a depth of 0.025-0.030 to give a Rockwell hardness figure of C58-C62. The $\frac{1}{8}$ in diameter, En 32

tappet adjusting screws are cyanide hardened, to a depth of 0.010 in, and their ball ends are polished. Both rocker shafts are of T26 steel tube and their inside and outside diameters are 0.310-0.315 in and 0.6233-0.6243 in respectively. Unbrako socket plugs are screwed into the ends of the shafts to seal them. Each shaft is carried on seven D.T.D. 424 aluminium die cast rocker pedestals, this material being employed in order partly to compensate for differential rates of expansion which might cause variations in tappet clearance. The distance between the valve stem axes of Nos. 1 and 2, 3 and 4, and 5 and 6 cylinders is 4.3 in, while between those of Nos. 2 and 3, and 4 and 5 it is 4.6 in. Other data relating to valve gear is given in the table.

Cylinder head, manifolds and carburetors

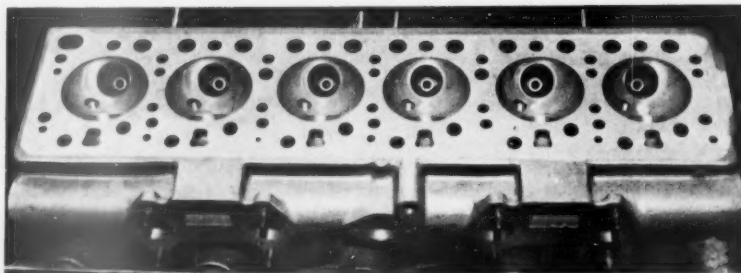
A B.S. 1452 grade 14 cast iron cylinder head is employed. The overall depth of the casting is 3.770-3.780 in and its overall width is 10.55 in. It is held down by fourteen $\frac{1}{4}$ in, B.S.F. bolts, of En 100T, on to a copper and asbestos gasket, the compressed thickness of which is 0.048-0.052 in. A Cooper EB cork washer forms the seal between the upper face of the head and the rocker cover. This cover is unusual in that it encloses both rocker assemblies, and is made from two 20 S.W.G. pressed steel shells which are placed one inside the other and separated by felt strips. It has rubber

grommetted holes in its top face, the grommets being fitted to seal round the tubes in which are housed the sparking plugs.

These tubes are vertically positioned, and each has a separate thimble end welded to its base; however, in future, it is intended that the ends of the tubes shall be spun over to form them to this shape. The whole assembly is a push fit in a socket in the top of the cylinder head, a copper washer being interposed between the thimble-end and the base of the socket. In the bottom of this socket is the sparking plug boss, and the plug is screwed down on to the usual copper and asbestos washer on the pierced thimble-end, and this prevents the tube from working loose as a result of vibration. A lip is formed round the top of the socket so that when the plugs have been removed, oil is prevented from running down through the sparking plug bosses into the cylinders, and causing a hydraulic lock when the engine is started again. The top of the tube is closed by a mushroom-shaped, plastics fitting, the stem of which is hollow and contains the plug lead and connection.

Hemispherical combustion chambers are incorporated in the head and each pair of valves is set, in a transverse plane, at an included angle of 70 deg. The axes of the inlet ports are turned through an angle of approximately 45 deg from the valve seats and the radius of curvature is 1.45 in, while the exhaust ports turn through approximately the same angle, but with a radius of curvature of 1.2 in. The water jacket is cored all round the exhaust valve guide housing, but only partly round that of the relatively cool inlet guide. Two separate cast iron exhaust manifolds are employed, each having three ports, and they are fitted in conjunction with Cemjo joint washers.

On the inlet side, an inlet rail or gallery is cast in the head and is joined by the six separate ports from the combustion chambers. The cast aluminium inlet manifold, on the other hand, has four branches which, of course, feed into the rail midway between Nos. 1 and 2, and 2 and 3 ports at the front, and 4 and 5, and 5 and 6 at the rear. Klingerit washers are employed between the manifold and the head. The front and rear portions of the rail are separated by a



Hemispherical combustion chambers are employed in this unit

$\frac{1}{4}$ in thick plate, in which is a $\frac{1}{2}$ in diameter hole that forms the balance aperture. A set bolt secures this plate perpendicularly to another which is bolted to the side of the cylinder head. Relative rotation between these two plates is prevented by a $\frac{1}{4}$ in diameter dowel. Below the inlet riser the manifold is water jacketed, and a fuel drain is incorporated at the lowest point. The crankcase breather system is connected to the induction air cleaner and silencer by means of a pipe from the oil filler neck on top of the rocker cover; and, on the right-hand side of the crankcase, an A.C. Delco air cleaner-breather unit is fitted.

With the single carburettor installation, a Stromberg DAV36 unit is mounted on an Incemjo insulating block, $\frac{1}{4}$ in thick, on the manifold. The choke diameter is $1\frac{1}{8}$ in, and the jet sizes are as follows: main (metering) jet 0.060 in, slow running (by-pass) jet 0.054 in, needle seating 0.100 in. When twin carburettors are fitted, they are the Stromberg DAA36 units, with a choke diameter of $1\frac{1}{8}$ in. The main (metering) jet is 0.058 in, and the power (by-pass) jet is 0.054 in. The needle seating is 0.100 in.

An A.C. Delco, series UE lift pump supplies fuel at a pressure of $1\frac{1}{2}$ - $2\frac{1}{2}$ lb/in². Fine gauze filters are fitted in the outlet from the fuel tank, the glass bowl on the fuel pump and in the carburettor union banjo bolt. The capacity of the tank is 16 gallons, and this includes $1\frac{1}{2}$ gallons reserve.

Water pump and cooling system

A fabric-reinforced rubber V-belt, with an included angle of 40 deg, transmits the drive to a cast iron pulley for the fan and water pump, which it drives at 1.105 times engine speed. Its overall dimensions are 0.390 in wide by 0.325 in thick. The pulley, together with the 14 in diameter, 4-bladed, 14 S.W.G. pressed steel fan, is spigoted and bolted on to a flanged boss pressed on to the 0.6262-0.6267 in diameter front end of the pump spindle. Positive location is effected by means of a grub screw inserted radially into the boss and engaging in a hole in the spindle. The assembly is statically balanced to within 7 inch drams.

A sealed tubular-type ball bearing, inserted from the front into the nose piece of the pump, carries the spindle. Location is effected by means of a dowel ended bolt which is screwed radially into the nose piece and which projects into a hole in the outer race. The inner races are formed by grooves in the spindle and their centres are spaced 1.06 in apart. In the space between the bearing and the water seal, no thrower ring is fitted but a drainage hole is drilled in its base.

The spring loaded water seal is

supplied by Super Oil Seals and Gaskets Ltd. It seats in a cup-shaped recess in the front of the rotor and has a moulded-in carbon thrust ring which bears against the front wall of the pump. This wall is part of the nose piece which is spigoted and bolted to the main body of the pump. Both the nose piece and the pump are of D.T.D. 424 and are anodized as a protection against corrosion.

Pressed on to the rear end of the spindle is a 2.95 in diameter stamped brass rotor. It is of the semi-shrouded type and discharges into a volute casing, which delivers coolant into a tube along the left-hand (exhaust) side of the top of the cylinder block jacket. Holes in the top of this tube direct the coolant up through cored ducts under the exhaust ports. From here, the coolant flows along the head and out past the thermostat to the radiator.

There is also a secondary circuit. In this the water passes out, through a thimble fitting pressed into the side of the cylinder head, into the water jacket, round the inlet manifold, and thence through another pipe back to the pump inlet. A rubber sealing ring round the shouldered outer end of the thimble is compressed by the manifold jacket cover plate when the assembly is bolted on to the head. The radiator is of the gilled tube type with brass tubes, and copper gills spaced $\frac{1}{4}$ in apart. It has a frontal area of 294 in², and four rows of tubes are accommo-

diameter for a length of $\frac{11}{16}$ in, below a point $\frac{1}{16}$ in from its slotted upper end.

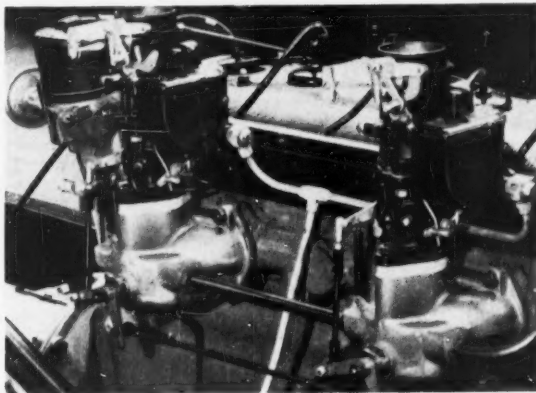
The En 32, integral spiral gear and spindle, by means of which the pump is driven from the camshaft, is tongued at its lower end. It spigots into the pump casing and the tongue registers in the slot in the top end of the pump spindle already described. Accurate alignment of the two spindles has been ensured by spigoting the pump casing into its boss in the crankcase; the spigot hole in this boss is, of course, accurately aligned with the housing for the plain bearing which supports the upper end of this spindle and its spiral gear. In this bearing, the spindle diameter is 0.5985-0.5905 in, but between the bearing and the pump its diameter is 0.562 in.

A somewhat unusual upper bearing arrangement has been adopted, so that replacement may be easily effected. The journal loads are taken by a Clevite bush, $1\frac{1}{2}$ in long, pressed into a cast iron bush which takes the thrust from the spiral gear immediately above it. This bearing assembly is located in the boss by a conical ended set screw which engages in a tapered hole in the side of the cast iron bush. This set screw, which is inserted from outside the crankcase, is secured by a lock-nut.

The upper end of the gear is shouldered and bears against a phosphor bronze thrust ring located in a recess machined round the inner

periphery at the base of the hollow turret casting that carries the distributor unit. This turret casting is of aluminium and is bolted and spigoted to a boss on the side of the crankcase. The drive for the distributor unit is formed by a $\frac{1}{8}$ in diameter, EN 8Q forged spindle, both ends of which are cupped. Its lower cupped end is tongued and registers in a slot in the top of the spiral gear. Spigoted into the cup at the upper end is the distributor spindle. A pin through the two components transmits the drive.

Oil is drawn through a floating pick-up in the pressed steel sump, which has a capacity of 10 pints. It discharges from the pump through a duct in the casing into a chamber formed by coring the sump casting between the bosses for the pump spigot and the pump bearing. This arrangement, of course, provides for the lubrication of the upper and lower bearings of the pump drive spindle. A small groove machined along the bore of the upper bearing housing directs a jet of oil on to the spiral gear. The positive supply of oil to the upper bearing is a noteworthy feature of this unit; in many designs this bearing is inadequately lubricated and in some cases it tends to squeak. From this chamber, which surrounds the upper drive spindle, the lubricant is passed



The twin carburettor installation

dated in the block thickness of $2\frac{1}{2}$ in. An A.C. Delco pressure cap maintains the system at its operating pressure of 4 lb/in².

Oil pump and lubrication system

A Hobourn Eaton oil pump, driven at half engine speed, is bolted up to a boss inside the wall of the crankcase, near the rear. The bottom cover of the pump is positioned immediately above the sump oil level, but the suction port, which is cast integrally with this cover, is below this level. A short spindle, $4\frac{1}{2}$ in long by $\frac{1}{2}$ in diameter, is carried in the pump casing and pegged in the driving rotor. This spindle is waisted to 0.4980-0.4985 in

through the full flow filter and thence to the $\frac{1}{8}$ in diameter gallery. It then goes through $\frac{1}{8}$ in diameter passages between the crankshaft main journal and camshaft bearings. Ducts are drilled in the usual way in the crankshaft to carry oil from the main journals to the big ends. The arrangement of the adjustable relief valve which, when the engine is warm, blows off at 40 lb/in², and the layout of the ducts serving it, which are in the rear intermediate bearing web, may be seen in one of the scrap views shown in the illustration of the engine cross sections.

The transverse drilling in the front wall of the crankcase is joined by two more holes. One of these carries oil into an axial drilling in the spindle of the jockey sprocket for the timing chain, and thence through a radial hole to lubricate the bearing surfaces. The other serves the thimble jet, screwed into the front wall of the crankcase, which directs oil into the meshing point between the timing chain and the drive sprocket.

An intermittent feed to the rocker

gear is taken from No. 2 camshaft bearing, in which a flat is machined in such a position on the journal that once every revolution the oil feed to the bearing is connected with a vertical drilling up through the crankcase, cylinder head and No. 3 pedestal, whence the oil passes into a hole drilled diametrically through the hollow inlet rocker shaft. A banjo bolt registers in the upper end of this hole to locate it and to carry the oil through an external pipe to a similar banjo connection on No. 3 pedestal of the exhaust shaft.

From the interior of both shafts, radial drillings serve the rocker bushes. The bores of each of the bushes are grooved and drilled to pass oil through a hole in the rocker, which communicates with an annular space round the waisted, tappet adjusting screw. From this space, a radial and an axial drilling in the screw pass the oil down to lubricate the spherical contact faces between it and the push rod end. Lubrication of the other end of each rocker is provided for by a $\frac{1}{2}$ in diameter hole drilled, from a point

between the upper and lower flanges of the I-section arm, radially through the boss and into the oil groove in the bush. The valves are inclined at such an angle that it is not necessary to take any special precautions to prevent oil running down their stems.

Electrical equipment

Lucas 12 volt electrical equipment is used throughout. A GTW11A battery of 64 amp-hr capacity is employed. It is served by a C45 PV5 dynamo operating in conjunction with an RB 106/1 voltage regulator and cut-out. The system is protected by a SF6 fuse unit. An M45G starter motor with a 10-tooth pinion engages the 142-tooth ring gear. The contact breaker and distributor unit is the DMX 6A. A centrifugal and vacuum operated advance and retard mechanism is incorporated, and supply is from a B12 coil. Champion N8B, 14 mm, long reach plugs are fitted. Other electrical equipment includes PF770 head lamps, 464 tail, stop and reversing lamps, dual tone WT614 horns and SF80 trafficators.

SWarf SEPARATION

THE present high cost of non-ferrous materials makes it essential that, for the sake of economy, swarf reclamation shall be as complete as possible. In the case of organizations that operate their own foundries, the reclaimed swarf must be completely free from ferrous materials, while if the swarf is to be sold as scrap its value is greatly increased if it is free from iron contamination.

In the machine shop there is usually the possibility that some mixing of ferrous and non-ferrous swarf will occur. Fortunately, the swarf can readily be separated in a magnetic separator of the drum type. This machine has horizontal magnet poles

so that as the swarf is fed on to the drum the iron particles form up in rows across the drum. During the rotation of the drum the non-ferrous material passes forward but the ferrous material clings to the drum along the lines of the magnetic poles.

Several ribs are fitted to the drum. As they move forward the non-ferrous material is pushed away from the magnetic pole and in jumping to the next magnet bar it turns completely over. By this means any non-ferrous material that may have been trapped is released. In practice a drum would normally have six sets of magnet bars, so that by the time the swarf has passed over the drum there is complete

separation of ferrous from non-ferrous material.

Two disadvantages of this type of equipment have been the high feeding height and the necessity for having cams at the back of the feed tray to jolt down the material, which does not flow freely. With this arrangement there was excessive noise and continual wear on the cams. To overcome these drawbacks The Magnetic Equipment Co. Ltd., Lake Works, Portchester, Hampshire, have developed the Magco feeder for use with their magnetic separators. This feeder has no moving parts. It has lowered the feeding height to 40 in and reduced the noise level to a very low value. (2051)

HARD CHROMIUM PLATING OF ALUMINIUM

AN article entitled "Hard Chromium Plating of Aluminium," by E. Meyer-Rassler, has been published in *Metal*, Vol. 6, No. 17/18. The author states that the resistance of aluminium and aluminium alloys to mechanical wear can be greatly improved by hard chromium plating. This process is especially suitable for engine cylinders, because the chromium possesses good corrosion resistance and radiation properties, as well as offering a low frictional resistance. To obtain good adherence between the aluminium and the chromium plate, however, a pre-treatment of the light alloy is regarded as essential.

Methods of pre-treatment may be chemical or mechanical, and aim at

removing the aluminium oxide skin, and producing a roughened substructure to which the chromium can be firmly attached. Etching processes using an alkaline zincate solution, and then a nickel chloride solution have proved satisfactory. Wet sand blasting is not suitable as it leads to inferior adhesion of the plate.

The plating process can be performed in baths of the same composition as those used for plating steel and iron. Grinding and honing may be used for finishing. In the production of chromium plated aluminium cylinders, a chromium layer 0.10-0.15 mm thick has practically no effect on the thermal conductivity of the unit. Therefore, before plating, the bore size is machined

to about twice the plate thickness, or 0.2 mm. oversize, so that the nominal diameter will be obtained when the process is complete. Experiments are necessary to determine the best plating arrangement for each type of cylinder.

To give better oil-retaining properties to the smooth chromium plated surface, pores and indentations may be produced by chemical, electro-chemical, galvanic or mechanical means. Honing is minimized by keeping the thickness of the plate to 0.10 mm. In series production, cylinders are classified in variations of 0.005 mm in size, so that pistons may be selectively assembled. During service, used cylinders may be stripped of chrome and replated. *M.I.R.A. Abstract No. 6350.*

LOW-PRESSURE LAMINATES

Physical Properties: Design Considerations: Moulding Methods: Resin-impregnated Asbestos-fibre and Glass-fibre Materials

Richard Wood

THE use of reinforced plastics in the automobile industry is not new and for a number of years materials consisting of a filler bonded with resin, in many cases under conditions of heat and pressure, have been used experimentally for body construction. More recently, with developments in the resin field, attention has been devoted to the study of suitable reinforcing materials and a considerable amount of development work has been carried out with fillers of asbestos, glass-fibre and a number of synthetic fibres such as nylon and terylene. Of these materials, the fillers that would appear most suitable for body construction are asbestos-fibre and glass-fibre. The nature and application of these two types of plastics materials have been discussed in *Automobile Engineer* for June 1952. Although certain of these materials have been available for some time and a number of experimental bodies have been built, it is possible that many of the advantages to be gained from their use have not, hitherto, been apparent, due to the fact that aluminium and steel have proved adequate, both economically and in production, for the construction of the more conventional types of body.

Properties

By reference to Table I, in which are set out some of the physical properties of reinforced laminates and traditional materials, comparisons are possible. At a first glance the specific properties of the reinforced plastics would appear to be low and it would seem doubtful if their use as structural materials would be advantageous. Each of these materials, however, possesses a quite distinct range in which it can be used efficiently and over the lower

Reinforced plastics laminates have received considerable attention during the past few years as possible alternatives to traditional materials for automobile body construction. As with many new materials, the reception of plastics has, in some cases, been over-enthusiastic, whilst in others, their application has been viewed with undue pessimism.

With a clear appreciation of their characteristics, these materials should be capable of selection and should prove a useful, if not revolutionary, addition to the constructional materials at the disposal of the automobile engineer.

ranges of loading, high efficiency structures from a viewpoint of strength: weight ratio can be designed.

In consideration of any design, torsional stiffness is of considerable importance and whilst these materials may not show to advantage in all cases, it has been found that due to a high strength:weight ratio, adequate stiffness can be obtained by designing for the use of the materials and not merely endeavouring to reproduce conventional structures without modification. Other advantages to be taken into consideration are low final cost which is particularly marked in the case of small quantity production due to simplicity of the tooling required and the fact that by careful design complex structures can invariably be produced as a single moulding, thus eliminating assembly time.

Among other advantages are good heat and sound insulation and an exceptionally high shock-absorbing property. This property is particularly important in an automobile body and greatly assists in localizing any damage in the case of a minor accident.

An additional factor of importance in

assessing the suitability of plastics as structural materials, is that contours which are difficult and expensive to produce in metal can be moulded with ease. In fact, it would seem that the development of plastics is opportune in view of the tendency toward one-piece bodies and the introduction, on the part of the stylist, of re-entrant curvature. From the point of view of road performance, plastics offer advantages in weight-saving, not only from their high strength:weight properties, but also due to the fact that as the materials consist of a number of layers of reinforcement, the strength of a structure can be controlled readily by the variation of the number of plies over a given area. In this way a body can be given high strength where stresses are high and yet need not pay a weight penalty when stresses are low.

Laminates also offer distinct advantages inasmuch that they are corrosion- and water-resistant and do not require extensive pre-finishing treatment. The operation is largely dependent upon the method of moulding adopted but, in many cases with a correct moulding technique and the application of a suitable mould-release agent, the treatment can be reduced to a minimum if not dispensed with entirely.

In the field of glass-fibre reinforcement, considerable work is being carried out with moulded-in colour and bodies have been produced which do not require painting. The process, however, is not yet entirely satisfactory from a commercial viewpoint due to the fact that the colours available are slightly transparent and the weave of the reinforcement can be seen. Although not without advantages for certain parts such as, perhaps, fascia boards, sun-visors and internal trims, the development of opaque colours is a

TABLE I. STRENGTH OF SOME LAMINATED PLASTICS AND METALS

	Specific gravity	Tensile strength (lb/in ²)	Compressive strength (lb/in ²)	Shear strength (lb/in ²)	E (lb/in ² × 10 ⁶)	Specific	
						Tensile strength (lb/in ²)	E (lb/in ² × 10 ⁶)
Durestos	1.27	15,250	13,000	5,000	2.0	12,000	1.57
Gordon Aerolite	1.43	70,000	34,000	2,000	7.0	49,000	4.89
Glass fibre (polyester)	1.82	43,000	42,000	8,000	3.0	23,600	1.65
Kraft paper (phenolic)	1.47	30,400	25,800	4,500	2.1	20,700	1.43
Duralumin	2.8	60,000	60,000	36,000	10.0	21,400	3.57
Mild steel sheet	7.8	45,000	45,000	22,000	29.0	5,800	3.7

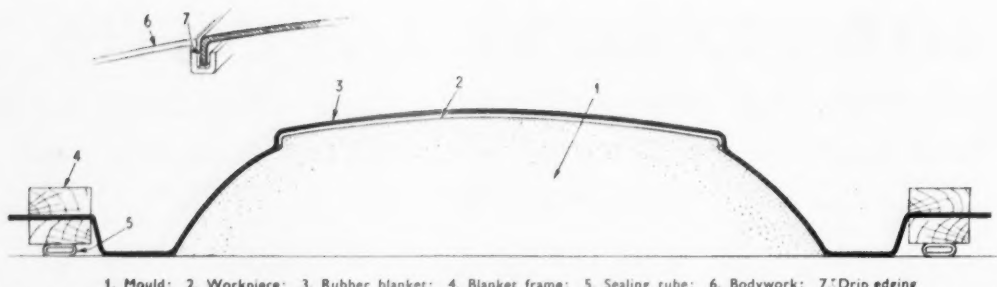


Fig. 1. Semi-diagrammatic view through a male-type vacuum-mould showing one method of moulding an engine access panel. Detail shows the method of moulding drip-edging

necessity if the technique is to be used commercially to replace cellulose and synthetic enamel for body exteriors.

On the debit side, plastics are expensive when compared with traditional materials of construction. Raw material costs, although showing some reduction, are high and production processes at present available, for large mouldings in particular, are time consuming. Output of raw materials, although increasing, is still comparatively small but with a greater demand it can be expected that prices will be further reduced.

Asbestos-fibre reinforcement

As previously described, asbestos fibre reinforced plastics are available commercially in the form of pliable felts which are impregnated with phenol-formaldehyde resin. This resin is thermo-setting and is of a two-stage type, that is, its cure takes place in two phases under the influence of heat and pressure. During manufacture the resin is brought to the first stage, so that after moulding, further application of heat and pressure brings about the final cure. It is of some interest to note, however, that although in production the application of heat and pressure is required to cure the resin thoroughly, the store life of the felts is not infinite due to a slow polymerization of the resin which starts and continues slowly from the time of impregnation.

A characteristic of the resin is that its cure is accompanied by a process of chemical condensation with the liberation of water which, to permit the resin to set, must be removed during the curing process. Until recently, this consideration has been one of the greatest obstacles to the use of phenolic-impregnated laminates for the manufacture of structures such as car bodies due to the prohibitive cost of the large, heated pressure-moulds necessary to the dispersion of the products of condensation. An additional problem in the use of phenolic-based felts, particularly in structures embodying a number of plies of the material and moulded without pressure, is that of the lifting of the plies due to liberated moisture.

Recently, to avoid the cost of large moulds, methods have been developed which largely overcome the necessity for the application of pressure. These

methods, which are known as vacuum- and no-pressure moulding, have permitted small quantity production of large structures to be carried out economically, and considerable experience has been gained in the handling and the application of the material.

During moulding it has been found that the asbestos-felts become soft and pliable as temperature is increased, and in this state very little pressure is required to consolidate them to each other and to the form of the mould. In this way, successful moulding can be achieved by ensuring that the folds are brought into intimate contact with one another and with the mould, and by continuously venting the mould during the curing period to disperse the products of condensation.

No-pressure process

The no-pressure process of moulding asbestos-phenolic material, which is based upon the foregoing, relies for its successful application upon a number of simple principles. Briefly, these principles are, the softening of the felts prior to laying-up, adequate consolidation of each layer, the application of a suitable adhesive between each layer, the provision of some form of internal restraint to prevent springback, and the provision of a suitable catalyst to enable the resin to cure at a sufficiently low temperature to minimize the vapour-pressure of the water given off.

Softening of the felts, it has been found, can be carried out conveniently by the application of warm water immediately prior to the commencement of the lay-up process. Consolidation of the felts is carried out in two stages, firstly with a narrow roller by which a heavy, local pressure can be applied, and secondly, by the use of a wide rubber squeegee to iron out variations in the thickness of the felts and to enable excess water to be removed. The application of an adhesive between each layer of felt to increase the homogeneity of the laminate is necessary because as the felts are cured without the pressure, adhesion between each layer is poor.

The need for providing some means of restraining the felts internally and the presence of a catalyst to reduce the curing temperature of the resin can be met by the use of a suitable adhesive. For this purpose a modified resorcinol-formaldehyde glue has been developed.

This glue is available in proprietary form as Aerolite 185*, which is activated for use by the addition of a small quantity (one-fifth by weight) of para-formaldehyde and when the glue has been mixed 1-2 per cent of furfuryl-alcohol is added. The addition of the alcohol, which is an excellent wetting agent, facilitates the dispersion of the glue into the fibres of the asbestos felt. Additionally, the modified resin gives considerably greater shrinking properties than the unmodified resin, and, in this way, serves to hold the layers of felt together during the curing process. The third advantage accruing from the use of modified Aerolite 185 resin is that when it is mixed with phenolic-resin, it has the effect of reducing considerably the temperature required for curing.

The actual lay-up of a component is comparatively simple and can be carried out after practice by a semi-skilled operator. The felt, after being cut to size—a standard bandsaw is suitable for cutting the felt—is damped on both sides with water at approximately 70 deg C. The felts are then laid in position on the mould and the first layer is compressed to the form of the mould by the use of the smaller of the two rollers. The second roller is then brought into use to expel excess water, and the outer surface of the felt is coated with the modified resin. The second felt is then placed in position and the process is continued until a laminate of the required thickness has been built-up.

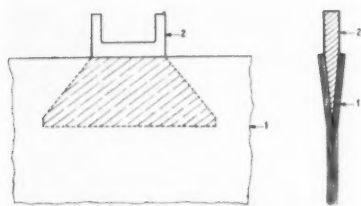
An important consideration in the tailoring of the felts is that the cuts are made in such a way that adjacent edges can be butt-jointed and overlapping edges avoided. It has been found that when the felts are wet it is comparatively simple to bring excess material into the form of a fold which can then be cut off close to the former.

Low-pressure or vacuum-moulding

For the larger-scale production of small parts, manufacturing costs can be reduced in many cases by the use of a low-pressure moulding technique. These techniques are many and varied and are based upon the application of a low pressure upon the surface of the work during the curing period.

In vacuum-moulding, atmospheric

*Manufactured by Aero Research Ltd., Duxford Cambridge.



1, Laminate; 2, Metal attachment point

Fig. 2. One method of incorporating metal attachment points in a laminate

pressure is used to consolidate the felts and to position the lay-up on the die or in the mould as the case may be. In its simplest form, the vacuum-moulding process consists of covering the mould and the lay-up with a thin rubber blanket, the underside of which is connected to a large-capacity vacuum pump. As the vacuum is drawn the differential pressure of the atmosphere conforms the blanket and the felt to the contour of the mould, whilst the vacuum-pump continuously removes the products of condensation. For the production of small parts, the operation is usually carried out on a specially-constructed vacuum table. To facilitate handling, the rubber blanket is stretched over a frame, the underside of which is provided with a resilient sealing strip to avoid loss of vacuum. For the production of large structures in which the depth of moulding is considerable, it is necessary to use a blanket which is contoured to the shape of the moulding. These blankets are usually made by the vulcanization of a number of shaped pieces and incorporate one or more connections on the inner surface for the attachment of the vacuum-line.

Mould materials

Both male and female moulds can be used for moulding asbestos-phenolic materials. Male moulds are, almost without exception, cheaper than female moulds, but the latter naturally

produce a superior and more consistent surface finish. Male moulds are particularly suitable for the production of prototypes as modifications and additions to the contour of the work can be incorporated readily and with the minimum of cost. Female moulds are preferable for quantity production and, as will be discussed later, are usually provided with means for heating the laminate to reduce curing time.

Materials suitable for moulds are many and various and include plaster, cement, and metals. The selection of material for moulds is mainly dependent upon the moulding process used and the size and the nature of the work. Where quantity requirements are small and not sufficient to necessitate the use of heated moulds, plaster can be used with success. For certain types of

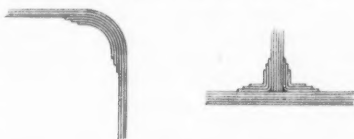
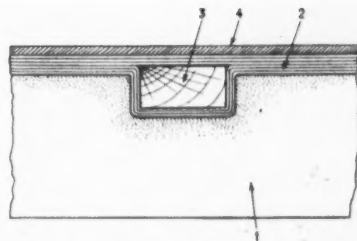


Fig. 3. A method of reinforcing bends and joints by additional plies of impregnated felt

work, low-melting-point alloy offers many advantages and can be used for the low-quantity production of small one-piece mouldings in which the form of the part precludes the withdrawal of the work from the mould.

Mould heating

In the simplest forms of mould, heating of the work to effect complete curing of the resin is by means of an oven in which the charged mould can be heated for a predetermined period. In some cases mould temperature can be maintained by means of induction-heating. Large moulds, particularly those of concrete, are usually heated by a number of resistance-type elements which are embodied in the mould during manufacture. They usually incorporate a series of thermocouples



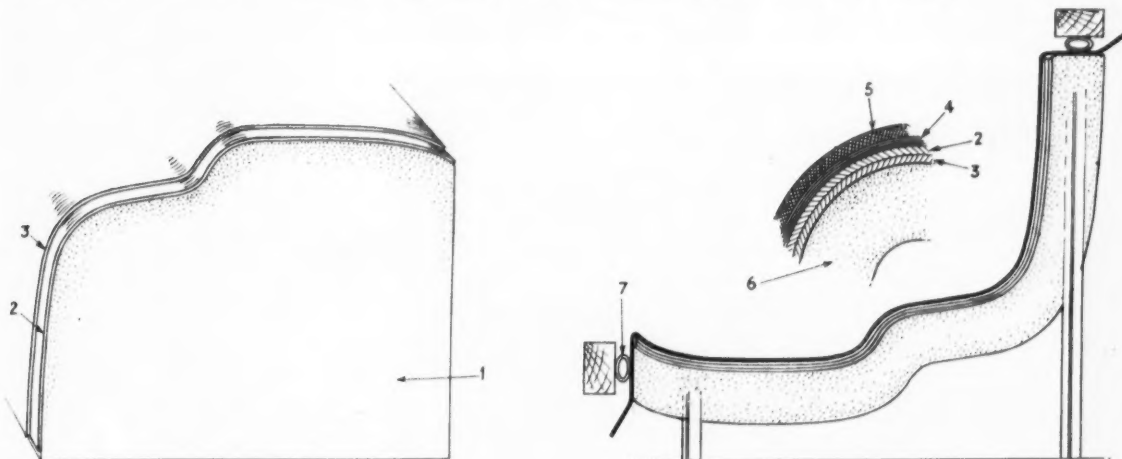
1, Mould; 2, Laminate; 3, Insert; 4, Rubber blanket

Fig. 4. Section through a male-type mould showing a method of embodying wood or metal inserts

to indicate the temperature conditions within the mould.

The vacuum-bag process has a wide range of application and can be used for the production of both large and small parts such as seats, facias, floor members, boot lids and bonnet lids. An example of one method of moulding a bonnet lid is shown semi-diagrammatically in Fig. 1. When used for quantity production of a number of similar parts upon male moulds, handling time can be reduced by curing a number of parts simultaneously upon the vacuum table. One factor of importance that must be taken into consideration when moulding a number of parts at the same time is the spacing of each mould or form block to ensure that the rubber blanket is not subject to excessive local stretching as the vacuum is applied. The resilience of the rubber blanket must also be taken into consideration in the design of tools for use in the vacuum process. Sharp corners should be avoided and ample radii and fairing on the lower edges of the tool should be provided to avoid undue stretching of the rubber.

Among advantages of this method of moulding is the ease with which stiffeners and inserts can be incorporated during the laying-up process. Although stiffeners and attachments of asbestos and other materials can be bonded successfully to the laminate



1, Model; 2, Intermediate liner; 3, Mould liner; 4, Laminate; 5, Rubber blanket; 6, Mould backing; 7, Sealing tube.

Fig. 5. (Left) Semi-diagrammatic section through a plaster model showing the position of the mould liner and the intermediate liner. (Right) Section through the completed mould with the rubber blanket sealed at its edges and under vacuum



Fig. 6. Chopped-strand mat. Scale 1:1

after it has been cured, where possible the attachment should be completely bonded-in and cured with the laminate. In this way superior strength is usually obtained and the insert, particularly if it is of metal, is not exposed to corrosion or rusting. The surface treatment of metallic inserts prior to bonding has received considerable attention recently and will be dealt with in more detail later in this article.

Mould liner

Recently the vacuum-bag method of moulding has been modified and for the production of large structures has been used in conjunction with a detachable mould liner. Moulding is carried out in a female mould, usually of concrete, under conditions of heat and pressure. In this process, the contour of the mould is obtained from an accurate model of the part which is used for the production of a phenolic-impregnated asbestos liner. This liner is detachable from within the mould itself and it will be appreciated that, as it is moulded from the model, its inner surface corresponds faithfully to the surface of the mould and this surface is, in turn, reproduced upon the outer surface of the part itself.

For the production of an accurate moulding, the use of an intermediate-liner is not essential as the layers of

felt that comprise the moulding could be positioned directly within the mould. However, the use of an intermediate liner, in which the felts can be laid-up before being loaded into the mould, ensures that the assembly cannot adhere to the surface of the mould during curing and, upon completion of the process, that the moulding can be removed without damage to itself or to the mould face. The complete mould, therefore, is composed of two parts: an inner lining of asbestos felt into which the intermediate liner is fitted, and an outer portion or shell of cement which stabilizes the asbestos-lining and serves as an abutment for sealing the edge of the rubber-bag that is used to compress the felts during the curing process.

It will be readily appreciated from the foregoing that the use of an intermediate liner ensures that in the event of adhesion between the work and the mould, the mould itself does not suffer damage and it is merely necessary to replace the liner.

The heat necessary to ensure complete curing of the moulding is obtained in this type of mould by resistance-type elements which are built into the mould during its manufacture. For the majority of work it is advantageous to heat the moulding from both sides during the cure and for this purpose a second series of heating elements woven on glasscloth blankets are placed on the inner surface of the felts after the assembly has been placed in the mould. Temperature conditions within the mould during the curing period are critical and to enable accurate control to be maintained during this period a number of thermocouples are usually embodied in the mould.

In equipment recently developed in the aircraft industry, a very large number of thermocouples were incorporated and to permit each thermocouple to be read at the required frequency, the temperatures are recorded automatically upon a chart.

Master-models can be made of a variety of materials, including plaster, wood and cement. For the majority of

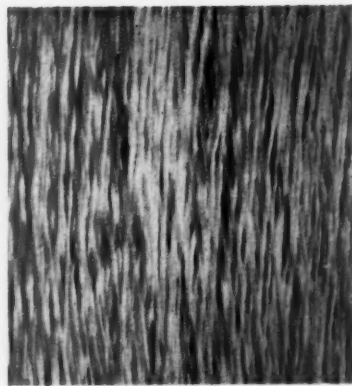


Fig. 7. Diamond mat. Scale 1:1

large work, plaster is usually the most convenient as it can be built-up on a wood or metal framework and can be finished to a smooth contour with ease. The contours of the master-model can be determined by the use of templates which are set-up at the required stations before plastering is commenced. Fairing of the model between template stations can then be carried out by the use of a spline in the usual way. The production of a smooth finish of accurate contour on the surface of the model is facilitated by the use, as a final covering, of a slow-setting plaster such as Keene's cement which permits work to be carried out on the mould for a period of some hours. It is well known that, if required, the setting period of the plaster can be extended by the addition of certain salts which are dissolved in the plaster while it is being mixed.

Intermediate liner

The next stage in the preparation of the complete mould is the production of the intermediate-liner. As previously mentioned, this liner is made of impregnated asbestos felt and is moulded over the model by the vacuum-bag process. To ensure release of the liner at the completion of the curing process, the surface of the model is treated with a parting agent—usually a grease containing a small percentage of silicones—before the felts are placed in position.

The mould itself consists of a number of layers of impregnated asbestos felt in which are located the heating elements and the thermocouples. The contour of the mould is obtained from the model in a similar manner to the intermediate liner. The investment of the mould in concrete to provide stability offers few problems, but difficulty has been experienced in preventing loss of vacuum in the mould during the curing process due to the porous nature of the concrete. To overcome this difficulty, it has been found advantageous to spray the outer surface of the mould with a vinyl emulsion to close the pores in the cement.

The operation of charging the mould can be performed by hand in the case



Fig. 8. Experimental body in glass-fibre/polyester resin with tubular steel reinforcement. Produced on an expendable plaster mould

of small mouldings and consists of laying each felt, in turn, in the correct position within the intermediate liner. Before being assembled, resin is applied to the surface of each felt. The charging of large moulds presents greater difficulty due to the problems of handling the prepared felts. Charging of these moulds, it has been found, can be greatly facilitated by the use of a loading-fixture over which each prepared felt can be draped in turn, and on which the assembly can be conveyed into the mould.

When the mould has been charged and the rubber vacuum-bag has been placed in position, it is sealed at the edges to prevent loss of vacuum. Provision for sealing usually takes the form of a rubber hose which is held by means of clamps against a smooth locating face on the edge of the concrete mould itself. The efficiency of the seal can be increased by making the hose endless and inflating it from the shop air line.

The average curing time for phenolic-asbestos felt varies widely. In the application described here, the curing period is approximately 6 hr in duration and is carried out in three stages. The first stage in the curing consists of raising the temperature to about 160 deg C at a rate of 70 deg/hr. In the second stage the temperature is maintained for a period of approximately 1 hr, after which the temperature is slowly reduced until the mould reaches room temperature.

It will be obvious from the foregoing brief review of moulding methods that the problem associated with the use of phenolic-impregnated asbestos felt material for the construction of automobile bodywork, except perhaps for the specialist coachbuilder, is the length of time required to complete the curing operation. Problems of tooling and material handling, although somewhat different from those asso-



Fig. 9. Singer Roadster body moulded in glass-fibre/polyester resin

ciated with more conventional methods of construction, should present little difficulty. From that aspect, it should be possible, with ingenuity, to speed-up the moulding process to the level required for medium production. The problem of curing, however, remains all important. Recently considerable development work has been carried out in this field and it has been found that the time required to cure a laminate of phenolic-impregnated asbestos can be reduced to approximately one-fifth the time normally required, by passage of an alternating current. This process, which at present is still in the development stage, has been termed shock-curing. The theory of the process is not yet fully understood but it is thought that the current may serve as a catalyst, or that polymerization is carried out more rapidly by an orientation of the molecules.

In the majority of the experiments carried out, specimens of laminate have been compressed between polished copper platens at a pressure of approximately 120 deg C. Under these conditions an alternating current

at a pressure of 50 V caused the temperature of the laminate to rise to between 160 deg and 200 deg C. In one of these tests a specimen of laminate was cured in a period of 20 min. The current, which in this case was 230 V, flowed for a period of from 5 to 10 min; the remainder of the time being taken in raising the laminate to the triggering temperature. As would be expected for a given material, the higher the voltage the more rapid the cure. In view of these developments it may well be that the solution to the problem of obtaining a rapid curing cycle lies in a modified form of shock-curing.

As a preliminary to reviewing the application of glass-fibre reinforced laminates in the automobile field it is advantageous to consider, briefly, the nature of the materials available as reinforcement and the properties of the various types of laminate that can be produced with them. Reinforced low-pressure laminates, as distinct from other fibre-filled plastics, are virtually tailor-made materials, inasmuch that their characteristics can be varied over quite wide limits to give properties suitable for a great variety of products.

In principle, all low-pressure laminates are produced by impregnating a number of layers of reinforcing material with a resin which is usually of a cold-setting type. Laminates produced in this way are analogous to a matrix such as reinforced concrete; each fibre being embedded in a tube acts as a reinforcing rod. While the primary purpose of the resin is to bond each of the individual fibres together, it does not contribute to any very great extent to the tensile strength of the laminate although, to some degree it does increase the stiffness of the structure.

As previously mentioned, one of the advantages of this form of construction is that the contours of the work are not limited in complexity by considerations of what can be produced economically under the press or drop-hammer, and the strength of the structure can be varied readily over any desired area by the number of plies of reinforcing material that are embodied in the lay-up.

For the preparation of low-pressure

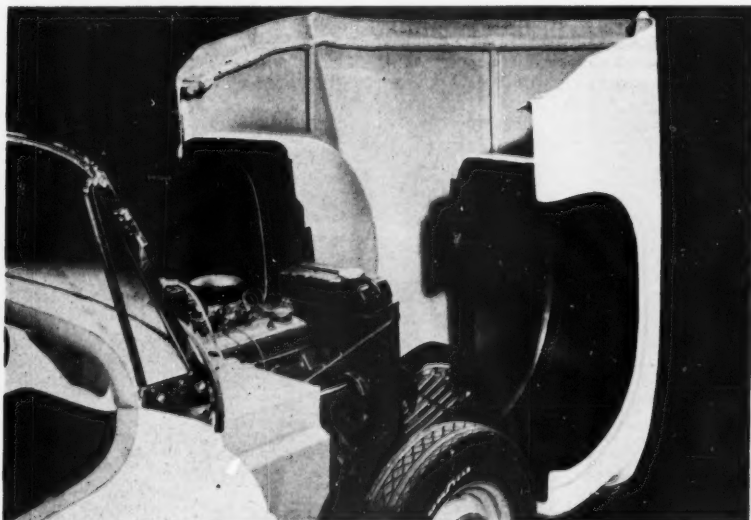


Fig. 10. The front portion of the Singer Roadster body in the open position, showing details of the tubular stiffening-members

laminates the use of a contact resin which will cure without the use of heat, or heat and pressure, is particularly attractive to the automobile body engineer. It eliminates the need for the heavy, heated presses and hydraulic systems which have been a necessity for the production of high-pressure laminates. The capability of working at low- or contact-pressures makes possible a wide variation in the method of laying-up the reinforcing plies and the comparative simplicity of the tooling required permits small numbers of a part to be produced economically and allows greater flexibility of design to be obtained.

Types of glass-fibre reinforcement

Glass-fibre reinforcement is available in a number of forms including cloth and woven and non-woven mats. Basically, all forms of reinforcement consist of a number of extremely fine glass filaments which are produced by melting glass nodules in an electric furnace and drawing the molten glass from holes in the base of the crucible by means of a high-speed winder. The filaments are then twisted and doubled to produce a yarn from which cloth can be woven or are wound parallel to produce a roving.

Cloths are basically of three types: a plain weave, in which each warp and weft thread passes over one thread and under the next, satin weaves which consist of four, six or eight shafts and in which each warp and weft passes under one and over a number of threads according to the particular type of cloth, and unidirectional cloths which have weak weft threads and strong warp threads to give maximum strength in one direction. The properties of cloth laminates vary widely with the weave of the cloth. In the main, laminates with the highest overall mechanical properties are those prepared from a large number of very thin plies or by the incorporation of thin plies in the outer surfaces. From this it will be apparent that the strength of laminates decreases with increase in



Fig. 11. A view of the Singer Roadster door showing the lock mounting and the method of obtaining edge-stiffness

the thickness of the individual plies. Impact strength, however, which is possibly of greater importance to the automobile body manufacturer, increases with the thickness of cloth.

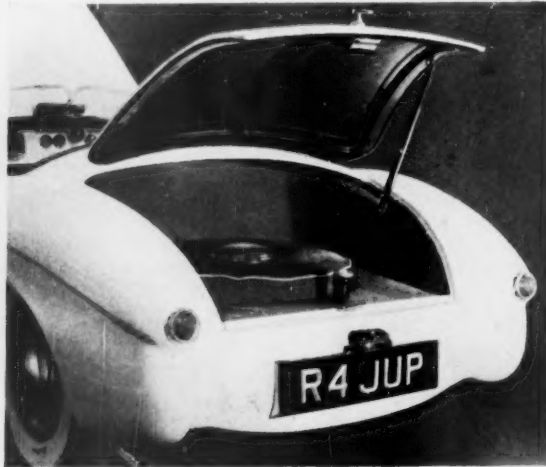
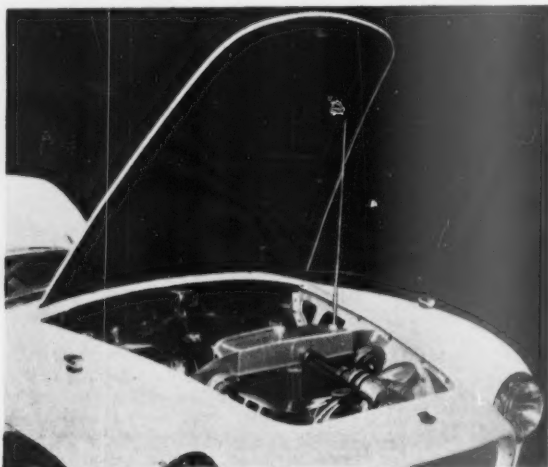
Briefly, therefore, the choice of fabric is dependent upon the design requirements. The advantages of each type of cloth may be summed up as follows:— plain-weave cloths offer advantages where uniformity of strength is required in two directions, usually at right angles, and where thicker cloths are used to facilitate deep drawing and the removal of air pockets when laminating by contact or vacuum methods. Unidirectional cloths find application where the requirements of the laminate are maximum strength in one direction, maximum strength:weight ratio, and where localized areas necessitate additional stiffness or impact resistance.

Long-shaft, satin-weave cloths are used for parts that require high strength in all directions combined with light weight. These cloths permit high-strength laminates to be produced at a lower cost and give a smooth surface finish.

Mats for use as reinforcement are of two types, those with non-orientated fibres and those with fibres that are laid in a diamond pattern. Chopped-strand or non-orientated mats give good multi-directional strength properties and are somewhat cheaper than diamond mats of comparable thickness. A further form of reinforcement is roving which, as previously mentioned, consists of a number of untwisted filaments. This material can be used for local stiffening and can be chopped into short lengths for use in preform moulding, which will be referred to later in this article. Although the surface finish obtained is dependent largely upon the method of moulding and the filler adopted, the finish on mouldings can be improved materially by the use of a special surfacing-mat which is incorporated in the surface of the laminate. This mat consists of a thin sheet of randomly distributed fibres of relatively large diameter which hold a comparatively large percentage of resin and thus gives a smooth finish.

Flexibility of styling

As previously mentioned, the fact that low-pressure laminates can be produced on simple tools permits a flexibility of design to be achieved that is not possible with a pressed-steel body. Although one of the most important characteristics of reinforced plastics is their suitability for one-piece bodies, it is yet too early to forecast what form bodies will take. A number of one-piece bodies have been constructed but, in this country to date, development on these lines has been carried out in the main by the amateur builder. Basically, the production of a one-piece body, unless of exceptionally simple shape and completely devoid of all under-cuts, necessitates the use of



Figs. 12 and 13. The Jowett Jupiter body is a composite structure. Bonnet and boot lids are of moulded construction, stiffened internally with top-hat section ribs

an expendable male mould, or a split female mould.

In the experimental body, Fig. 8, now being constructed by the writer for the purpose of developing methods for producing a closed body of exceptionally light weight, the lines of the body were conceived as an ideal and no concessions were made to the problems of removing the body from the mould. For this reason it was necessary to lay-up the body shell on a plaster male mould that could be broken out when the curing of the resin was complete. For the production of more than one body of similar or equally difficult contour the use of a split female mould would overcome difficulties due to undercuts and areas of return curvature.

Prototypes

The expendable plaster male mould, however, has been used to some advantage commercially by manufacturers in the U.S.A. for the production of prototype bodies for the purpose of obtaining decisions on such questions as styling, seating capacity, colour and accessories. Hitherto, the medium used for this purpose has been a clay or plaster. These materials have many disadvantages, chief of which is their weight and consequent difficulty of transportation and the problem of obtaining an accurate representation of chromium-plating on parts such as bumpers and screen pillars. Glass-reinforced plastics, it has been found, is an almost ideal material for this purpose, as an accurate representation of proposed bodies can be produced cheaply and the high strength:weight ratio of the moulding permits the structure to be transported with ease. Moreover, little difficulty is encountered in plating and painting.

Corrosion

The high corrosion-resisting and water-resisting properties of glass/polyester laminates is of particular value in automobile bodies, especially for those exported to maritime countries and localities where high humidity conditions accelerate corrosion and oxidization of alloy and steel bodies. Furthermore, the excellent thermal-insulating properties of glass/resin laminates is of marked value in countries having extremes of climate.

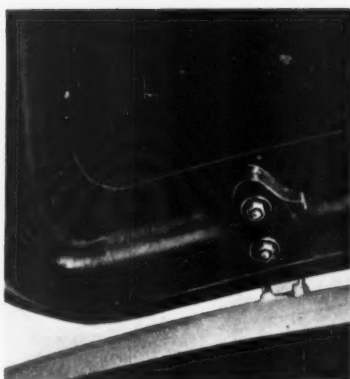


Fig. 14. Close-up view of the Jowett boot lid showing details at one of the hinge attachment points

As will be appreciated, the use of double-skinned panels, particularly for roof structures, materially increases the thermal efficiency and for hot climates the incorporation of a white-pigmented filler in the resin used for bonding the outer plies of the roof can effect a considerable reduction in the temperature of the interior of closed bodies.

Fire-proofing

An important consideration from the point of view of the automobile body engineer is the fire risk of non-metallic materials. Broadly, glass/polyester resin laminates do not support combustion although they will burn if subjected to the heat of a continuously applied flame. It will be appreciated that combustion is confined only to the resin content, the glass-fibre reinforcement being an inactive constituent. Complete fire-proofing of the laminate, however, can be achieved by the addition to the resin during the mixing operation of a small percentage of comparatively inexpensive materials. The fire-proofing material found most suitable, is a mixture consisting of 15 parts by weight of Cereclor 70* and 15 parts by weight of antimony oxide. In mixing these materials, the Cereclor is dissolved in the resin first and the antimony oxide ground into the mixture afterwards. The addition of fire-proofing materials

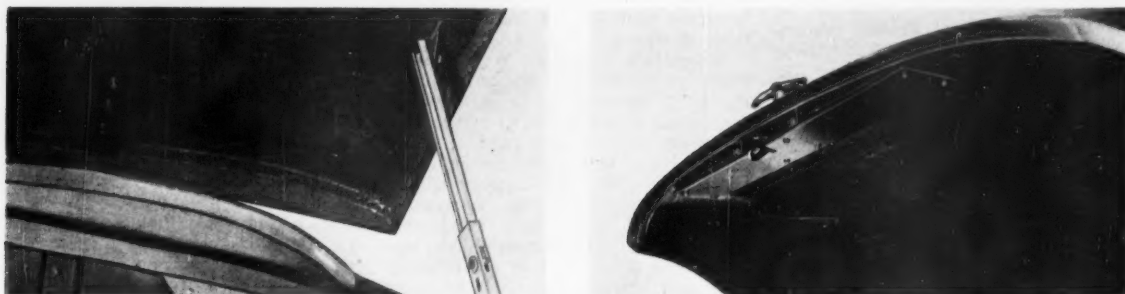
*I.C.I. Ltd., Plastics Division, Black Fan Road, Welwyn Garden City, Herts.

usually increases the setting time of the resin somewhat, but if the oxide is dry the increase is not considerable.

Design considerations

As previously mentioned, one of the most important advantages of glass-reinforced plastics is the ease with which complex curves can be produced, as compared with panel beating. If full advantage of the physical properties of the material is to be obtained, however, each application must be carefully studied in relation to traditional materials. Although possessing a high strength:weight ratio, the specific stiffness of glass-fibre resin laminates is comparatively low and reaches only about one-third that of many of the aluminium-alloys and steel. This fact largely governs the design of laminated structures and imposes upon the designer restrictions not present with metal. Stiffness requirements can be met largely in design by the avoidance of flat panels and the incorporation, where styling considerations permit, of double and return curvature. In the majority of bodies, however, there are usually a number of areas where the use of pronounced double curvature is precluded and the incorporation of stiffening ribs on the inner surface of the panel has been found satisfactory in reducing "oil canning" effects over these areas.

Although the majority of manufacturers in this country are following the development of reinforced plastics with interest, to date these materials have been used on very few production cars. At the recent Exhibition held at Earls Court, the Jowett Company showed a composite body in which the front wings, engine access panel and boot lid were of moulded construction and the Singer Motor Co. Ltd. showed a Roadster body moulded completely in resin-impregnated glass-fibre. The Singer body consists of four main units, the body aft of the scuttle and including the rear wings, the bonnet, radiator cowl and front wings, and off-side and near-side doors. The forward portion of the body is hinged at the front cross-member and can be raised to facilitate access to the engine and front suspension. The fascia also consists of a glass-fibre moulding. It is interesting to note that in this, the first production body to be made in this country, steel



Figs. 15 and 16. Details of the Jensen moulded boot lid. (Left) Stiffening at the hinge attachment points. (Right) Method of strengthening at the lock

stiffening members are incorporated in the mouldings. In the body itself the stiffening members are out of sight and can be seen only in the front, hinged portion when it is in the open position. The incorporation of metal or wood stiffeners at points of maximum stress concentration offers a number of advantages, especially in the development stages of a body. As moulding techniques become more advanced, however, it will be possible to eliminate metal and wood and to embody integral stiffening during the laying-up process.

An excellent method of producing light, stiff panels is the use of double-skinning which has been developed by the aircraft industry for the manufacture of radomes. This method is used in certain parts of the ultra-light experimental saloon body, previously mentioned. In principle this form of construction consists of two comparatively thin panels of laminate bonded to a core of low-density material which serves as a filler. A number of materials of low density have been found suitable, including Ozonote,* which is a proprietary form of foamed-rubber sheet, and Dufaylite,† an expanded phenolic-resin impregnated paper honeycomb. More recently an expanded aluminium-alloy foil honeycomb has been developed in the aircraft industry for the construction of floors and similar structures where light weight combined with stiffness is of paramount importance.

Another and interesting method of increasing the rigidity of double-skinned structures is the use of a liquid foam-filling, for which a number of materials have been developed. This form of construction is attractive owing to the simplicity of the process and because of the exceptionally low-density of the filler material which is prepared in liquid form and can, therefore, be used to fill a cavity of any shape. Of the foaming resins developed in this country, the Sebalkyd‡ compounds have been found suitable for use in structures of glass/polyester. The Sebalkyd resin foam is produced by the reaction of isocyanates. Carbon-dioxide is given off during the reaction and is retained in multiple cells as the resin sets. The density range of the foam, which can be controlled to suit the particular application, varies from about 1½ to 30 lb/ft³. The foaming reaction is exothermic in nature and considerable internal pressure is developed which, for certain shapes, necessitates the use of restraining tooling. For some applications the internal-pressure is advantageous and serves to hold the outer skins against the face of the tool and increases the surface adhesion of the core and the laminate. Although possessing inherent advantages for certain structures, the process is expensive and in consequence will possibly be confined to critical components and to stiffening areas in

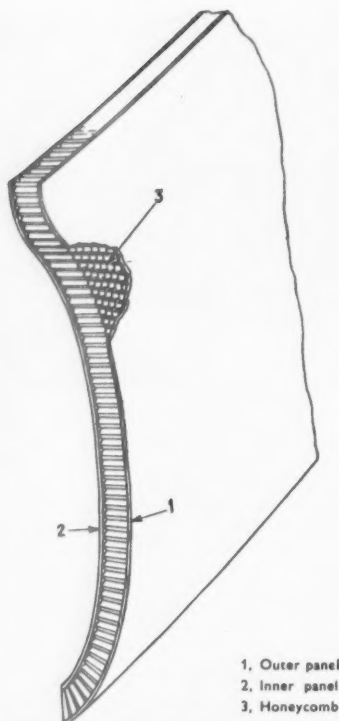


Fig. 17. Sectional view of a double-skinned door panel illustrating the use of a low-density honeycomb interlayer

high-performance, ultra-light body-work.

Attachment difficulties

One of the major problems in designing structures in glass-fibre laminate occurs where it is necessary to transmit load from a laminate into structures such as doors and into the chassis itself due to differing extensions when under load of the metal and the laminate. Much can be done to eliminate this difficulty at the design stage and recently adhesives have been developed for bonding metals to many of the newer laminated-plastics. In practice a number of methods have been evolved for joining stiffeners and similar parts. Broadly, the strength of the bond depends upon the absence, or at least the minimization, of flexing at the joint which may cause delamination of the plies. Where possible it is advantageous to cover the ends of the laminate with one or more plies of cloth, as shown in Fig. 19.

Where it is necessary to bolt laminates to traditional structures care should be exercised in the selection of the washer diameter to give the maximum bearing area possible upon the laminate, and consideration must be given to the edge-distance of all holes. The minimum edge-distance of holes should be at least twice the diameter of the bolt measured from the centre-line of the bolt to the edge of the panel. In addition, an increase in thickness is desirable at attachment points because of the low ductility of the glass-fibre reinforcement and lack

of relief at areas of stress concentration.

Secondary bonding operations are facilitated and the quality and load-bearing characteristics of the joint are materially increased by the incorporation in the final lay-up of a pull-layer, or extra ply of glass-fabric, which can be torn off immediately prior to the attachment of the stiffener or attachment. In this way, a uniformly roughened surface is presented which is entirely free of grease or fatty contamination which would reduce the efficiency of the bond. This technique eliminates sanding and overcomes the difficulty of producing a roughened surface in confined spaces and on the inner face of acutely curved panels.

Moulds

Both male, female and matched moulds can be used for moulding glass-fibre laminates. As already mentioned, male moulds are usually confined to the production of one-off and prototype work. Single moulds are the simplest type and can be made from a model of plaster, wood or metal. The mould can be either male or female, depending upon which surface of the work the finish is required. For the production of a car body the inside of the moulding is not important and in consequence a female mould which gives a good finish on the outside is used. The material of the mould has very little bearing upon the finished moulding and any of the materials mentioned can be used providing the moulding face is coated liberally with a mould-release agent before laminating is commenced. This operation is all-important because the finish obtained on the surface of the mould is reproduced on the outside of the body. It will be appreciated that careful preparation of the mould surface can materially reduce the cost of finishing each body produced—a time-absorbing operation and one requiring the services of skilled operators.

Laying-up

When the surface of the mould has been prepared it is usual to apply, either with a brush or spray-gun, one or more coats of high accelerator-content resin in order to give the body a hard, smooth surface and to eliminate air-bubbles and pin-holes. The layers of cloth or mat are laid in the mould when the last coat of resin has begun to gel or, at least, within two or three hours of its setting. If a longer period is allowed, the adhesion between the surfacing coats and the laminate may be reduced. Impregnation can be carried out with a brush or with a spray gun as each layer of reinforcement is placed in position. As the ultimate strength of the laminate is dependent largely upon its homogeneity, air-bubbles and voids should be eliminated, as far as possible, by squeezing, either by hand or with a roller. When using a roller it is usual, to avoid pick-up of the laminate, to cover the surface with a sheet of cellophane or to sprinkle the surface liberally with

*Expanded Rubber Co. Ltd., Mitcham Road, Croydon, Surrey.

†Dufay Ltd., 14/16, Cockspur Street, London, S.W.1.

‡Scott Bader and Co. Ltd., Wollaston, Wellingborough, Northamptonshire.

finely divided chalk. Glass-fibre laminates have good stability properties, but the best results are obtained from balanced laminates in which the outer and inner layers consist of the same type and number of plies of reinforcement.

Vacuum-bag moulding

The rubber vacuum-bag method of moulding previously described, can be used for moulding glass-fibre parts and produces an excellent and homogeneous laminate. Moulds for use with the vacuum process can be either male or female and can be made of wood, metal, plaster or glass-fibre laminate. To accelerate the curing-time, the moulds may be heated internally by steam pipes or externally by means of infra-red lamps. The reinforcement can be impregnated before or after it has been laid up but when using chopped strand mat, it is usually more convenient to lay the material on the mould before applying the resin.

To protect the vacuum bag from the resin and to prevent sticking, a sheet of cellophane or polyvinyl acetate is interposed between the surface of the bag and the moulding. This sheet has the advantage of providing a comparatively smooth surface on the inside or the outside of the moulding as the case may be.

The vacuum-bag method of moulding, although somewhat more complicated than the no-pressure method, permits work to be carried out more quickly. This is due to the fact that the application of pressure ensures complete impregnation of the fibre without an interval for soaking.

Pre-forming

A further method of producing glass-fibre laminates is by pre-forming. This process, which has been developed primarily for the quantity production of comparatively small articles, consists of scattering short filaments of glass on

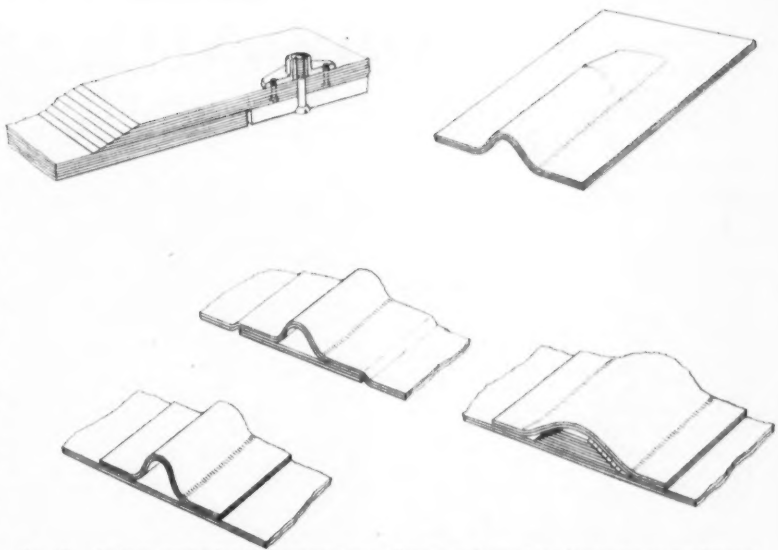


Fig. 18. Various methods of incorporating stiffeners and attaching laminates to traditional structures

to a fine-wire grille which corresponds in contour to the shape of the part to be produced. The underside of the grille is open to a large capacity air-pump which produces a stream of air through the grille sufficient to hold the filaments in position. While the filaments are being deposited upon the grille, a resinous binder is sprayed on to the material to hold the pre-form together and to permit it to be lifted off the grille when the desired thickness of mat has been built-up.

For moulding, the pre-form is placed in matched tools and impregnated with a hot-setting resin. The mould is then closed and subsequently the moulding is cured at a temperature of between 120 and 130 deg C.

Conclusions

In assessing the suitability of low-pressure laminates for automobile

bodywork, it would appear that as substitutes for traditional materials they offer outstanding advantages in weight-saving and in reducing body-maintenance costs. For short-run production or specialized work, the low tooling costs involved show to advantage, but for quantity production the inherent disadvantages are the comparatively lengthy laying-up process, the time required for curing and the difficulty of ensuring a consistently good surface finish without hand work. However, as more experience is gained, new and better moulding methods will, no doubt, be evolved to overcome these difficulties and to permit production in quantity. Moreover, large-scale production of the resins and other materials of construction may, conceivably, also lead to a reduction in their cost.

Laboratory Oven

AN easily transportable, low-temperature oven intended for laboratory work has recently been introduced by The General Electric Co. Ltd. It incorporates all the features required to permit checking and controlling movement of components during tests involving long periods of heat-treatment.

The oven is of standard double-case construction with ample heat insulation, and has a full-length door at the front. Forced air circulation is provided by a centrifugal fan mounted in the roof. Sheathed-wire elements are mounted on the side walls behind metal shields which protect the charge from direct radiation and at the same time form part of the air circulation system.

It has an internal capacity of 3 ft³ and is rated at 9 kW. Temperature is controlled by an expansion thermostat and the oven is provided with temperature indicator and recording instruments, door switch, pilot lamps, and thermo-couple connections. Maximum operating temperature is 350 deg C.

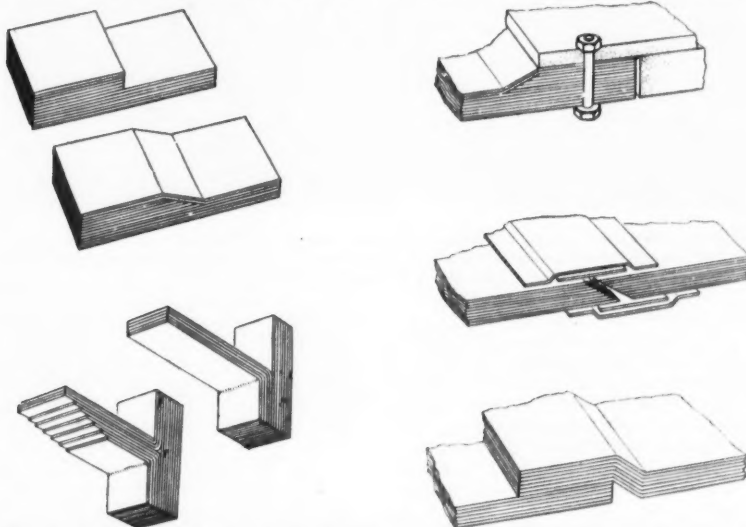


Fig. 19. Various methods of joining laminates and obviating delamination and ingress of moisture between the plies

CARBO-NITRIDING

A New Birlec Controlled-atmosphere Furnace for Light-case Work

TO an already wide range of heat-treatment equipment, Birlec Ltd., Birmingham, have now introduced a controlled-atmosphere mechanized batch furnace for carbo-nitriding light-case work. This new development is the outcome of a manufacturing agreement with the Dow Furnace Company, Detroit, Michigan, the first in the field in the U.S.A. with a furnace of this type. Combining the operating economics and advantages of large continuous furnaces with the flexibility of batch-type equipment, the furnace has become firmly established in the U.S.A. where over fifty are already in service. Leading American automobile manufacturers, in particular, have put down a number of installations.

Industry has long recognized the need for a versatile, controlled-atmosphere furnace capable of processing light-case work. Whilst suitable for deep-case work, the normal pit-type furnace has lacked the ability to control light-case depths and could not equal the surface finish obtained in the salt bath. This has meant that liquid cyaniding, with its high cost for labour, salt, maintenance and cleaning, has hitherto provided the only flexible means of producing a satisfactory light-case. A prototype furnace built for demonstration work at the Company's factory in Tyburn Road, Birmingham, has for some time now been producing a consistently high standard of case work on a wide variety of components.

Results obtained

The case produced by carbo-nitriding is similar to that obtained from the cyaniding process. Carbon and nitrogen from the enriching gases, butane and anhydrous ammonia, are absorbed at the steel surface and diffuse inwards to produce the case. The process is applicable to any steels which are at present cyanided, mild steel and carburizing steels in general as well as medium-carbon and low-alloy steels to which it is desired to impart a high surface hardness.

The temperatures used in gas carburizing are appreciably higher than those used for carbo-nitriding although the times taken to produce the same case depth are generally similar. In this respect, carbo-nitriding is comparable with salt-bath cyaniding. For example, to produce a case depth of 0.020 in by carbo-nitriding or cyaniding at 850 deg C, 2.25 hours at this temperature are necessary while 2.0 hours at 930 deg C are required to produce a similar depth of case by carburizing. The advantages of the carbo-nitriding process are that by using lower temperatures distortion of the components is greatly reduced and the many objectionable features

associated with salt-bath cyaniding, as mentioned earlier, are avoided.

Types of components already treated successfully in the demonstration furnace include mild steel wheel nuts, bicycle hub cones and chain wheels, as well as low-alloy gears, pinions and shafts. The resultant surface hardness has been generally between 800-850 D.P.N. on suitable steels.

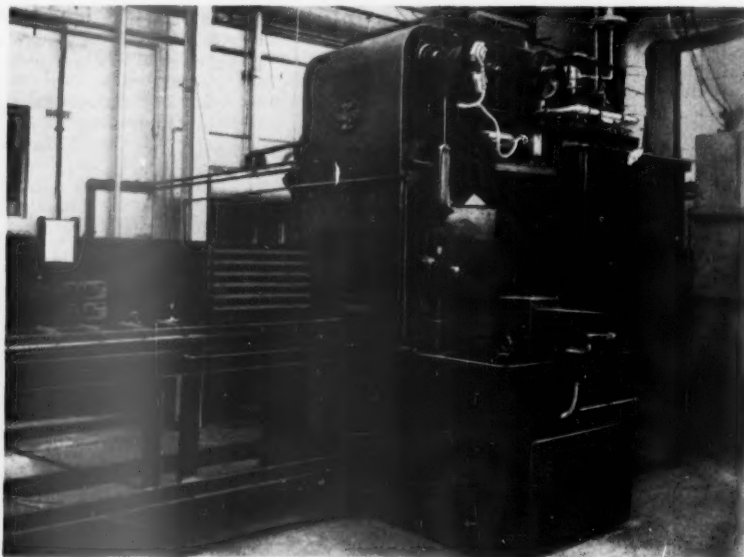
This new Birlec-Dow furnace is of the horizontal batch type, heated by vertical gas-fired radiant tubes. These operate, together with a refractory baffle, in conjunction with a powerful circulating fan placed in the roof of the chamber. The furnace is designed to ensure an unusually high uniformity of heating and case penetration on densely packed loads, coupled with an equal consistency of quenching, which is carried out without exposure of the charge to air. Complete freedom from scale and decarburizing is ensured, of course, by this controlled-atmosphere system of quenching.

atmosphere—to which ammonia and hydrocarbon are added for carbo-nitriding—which is fed directly into the furnace at the desired rate.

Another special feature of the furnace is in the provision of a high degree of thermal storage in the heating chamber by the use of baffle walls of refractory material placed between the radiant tubes and the charge. These walls serve not only to direct the high flow of atmosphere circulated by the fan through what may be a relatively densely packed charge, but also act as "heat capacitors" when a new charge enters the furnace.

Heat-capacitor effect

When a fresh load of work is put into a hot furnace there is a very rapid transfer of heat from the baffle walls to the circulating gases, which augments that given off by the radiant tubes. By the time the work in the furnace has reached its critical temperature that process is reversed, the walls by then

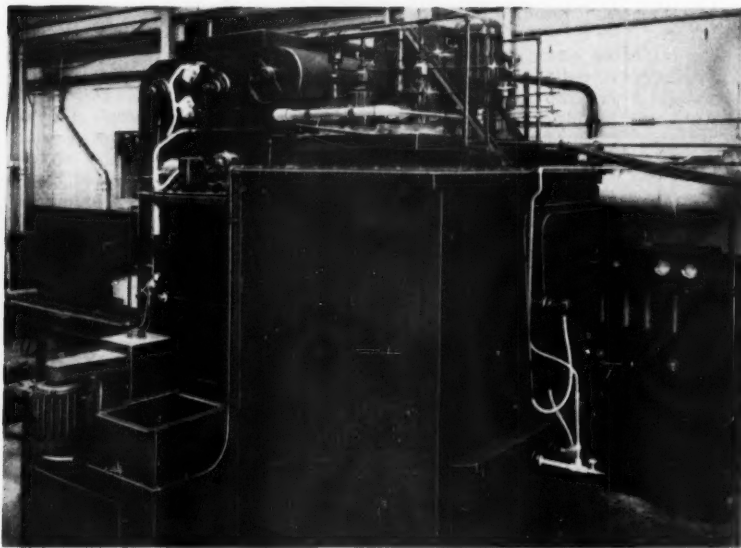


On the side of the vestibule chamber are power drives for vestibule and furnace doors. In the foreground is the quench tank with a motor-driven oil impeller. A charge container is shown on the loading conveyor

Self-contained generator

An important innovation is the incorporation of catalytic generators of endothermic type within the radiant tubes, which obviate the need for the more usual separate generator. This feature also enables the furnace to be used for clean-hardening medium steels in addition to carbo-nitriding and gas-carburizing. Mounted inside the U-shaped radiant tubes, these self-contained generators convert an air-town's gas mixture into the proper

are acting as heat absorbers. The walls are, therefore, able to store a considerable amount of heat produced by the radiant tubes, where temperature is maintained at a higher level by the action of the automatic control mechanism. The time required for the furnace to regain temperature after insertion of a new load is thus reduced. Through this heat capacitor effect the gradient in temperature through the load, as it approaches control temperature, is substantially less than that



Shown on the roof of the furnace are the motor drive for the circulating fan and the ends of two of the radiant tubes projecting above the sand seal

obtained when a simple radiation heating system is employed.

The rated input of the furnace is such as to heat 500 lb gross per hour from room temperature to 815 deg C with the furnace at control temperature when the load is introduced. The actual output of case-hardened work varies, of course, with the net load obtainable, the temperature and the required depth of case, but loads up to 600 lb can be processed, and outputs are generally in the range of 100-400 lb per hour. If required, a larger capacity model with a chamber and elevator designed for holding two containers arranged side-by-side can be supplied.

Integral quench tank

The front of the furnace is sealed by an enclosed vestibule which is situated immediately above the integral oil-quench tank. In this quench tank a motor-driven impeller is employed

not merely to agitate the oil but to direct it at a high-rate of flow right through the charge immediately this is lowered into the tank, thus ensuring uniformity of quenching to full hardness. The quench oil is maintained at a suitable temperature by finned water-circulating pipes housed within the quenching tank.

Inside the vestibule chamber is an elevator on which the charge is lowered directly into the quenching tank. The Birlec demonstration furnace has been provided with a two-tier elevator to enable a second charge to be held in the vestibule ready for placing in the furnace while the previous charge on the lower tier is immersed in the quench tank. The elevator has roller platforms serving, in effect, as an extension of the outside loading conveyor and of the rollers forming the base of the furnace chamber. It is supported on chains and is provided with an electric

motor drive. After immersion in the quench, the elevator is raised and the charge may then be withdrawn through the vestibule door opening.

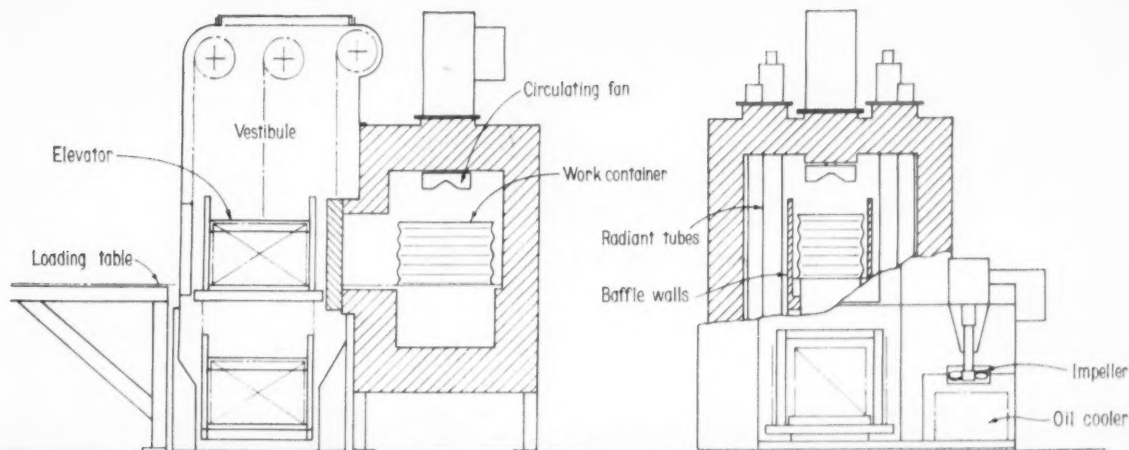
Construction

Occupying a space of approximately 9 ft 6 in by 9 ft, the demonstration furnace has an overall height of 10 ft 9 in; for removal of the radiant tubes a total head room of 15 ft is required. A mild steel casing is used for both the furnace and vestibule, the joints of which are welded gastight. The furnace walls are generously insulated to ensure a minimum of heat losses. Located between these walls and the heat capacitors are the four radiant tubes which are sealed at the points of entry through the furnace roof.

Also mounted in the roof of the furnace is the high-capacity circulating fan, the shaft of which is specially cooled. The furnace chamber has 1 ft 8 in of usable height above the nickel-chromium rollers and rails on which the load is supported and is arranged to take containers approximately 2 ft 5½ in long and 1 ft 7½ in wide. These containers have grid-type bottoms, to facilitate thorough circulation of the processing atmospheres and quenching mediums through the charge.

The front of the furnace, leading into the vestibule chamber, is sealed by an insulated door raised by a motorized chain drive. To give head room for the upper and lower elevator frames, the gas-tight vestibule projects above the furnace and has mounted on the sides driving motors and controls for the elevator and the furnace door lifting chains, and also the pneumatic cylinder for raising the vestibule door. The necessary power for operating this cylinder is drawn from the factory air lines. The vestibule door is provided with a small opening, having a sliding cover, to enable a hook to be inserted for sliding the work container in and out of the furnace. This operation may be performed by mechanical means, particularly on larger furnaces.

The height from floor level to the vestibule door is 40 in and the loading



General arrangement of the Birlec-Dow carbo-nitriding furnace

conveyor can be arranged parallel to or across the front of the vestibule or adapted in other ways to meet customers' own layout requirements. Extending below and on both sides of the vestibule chamber, the quench tank is provided with baffle plates to direct the flow of oil generated by the motor-driven impeller through the load when this is lowered into the tank. The layout of the furnace is such as to call for a minimum of skilled attention and it incorporates safety devices for shutting down in the event of power or gas failure.

Method of operation

Operation of the Birlec-Dow furnace follows well-established procedure. Air is kept out of the furnace by maintaining a positive pressure of endothermic atmosphere in the work chamber, and regulation of the hydrocarbon-ammonia addition is such that only sufficient carbon or nitrogen is provided for the process requirements. There is a prescribed cycle for starting, operating, and shutting down, and the furnace must be properly "conditioned" after a shut down and "burned out" at definite intervals to remove excess

carbon deposits from the furnace structure.

The use of an elevator in the vestibule chamber not only offers the advantage of enabling quenching to be carried out within the controlled atmosphere of the furnace, but also considerably reduces non-productive time. A continuous cycle of processing can be maintained by loading a work container and bringing this to the vestibule door during the time that a load is being processed in the furnace chamber. Before this is admitted to the vestibule chamber the elevator is brought to the down position by depressing the "elevator down" switch, which brings the upper platform in line with the vestibule door. The operator then admits air to the pneumatic cylinder that raises the vestibule door, and pushes the new charge waiting outside the vestibule on to the upper elevator platform. After the door is closed, air admitted to the vestibule during the operation is allowed to purge. The elevator is then raised to bring the lower platform in line with the furnace door in readiness to receive the charge being processed in the furnace chamber. When this charge is

ready for quenching, the furnace door is raised and the container is withdrawn on to the elevator by a hook inserted through the small slide-covered port in the vestibule door. The elevator is then lowered and the impeller motor switched on to agitate the quenching oil. As the elevator descends, lowering the treated charge into the oil, the new charge waiting on the upper elevator platform is brought into line with the furnace hearth and is pushed into the furnace chamber, after which the furnace door is closed. After approximately three minutes in the quench tank, the elevator is raised and the finished charge, after a sufficient interval for draining, is withdrawn from the vestibule. The loading cycle can then be repeated.

The experience gained in the U.S.A. and by Birlec with their demonstration furnace has not only confirmed the high efficiency of this type of furnace in practice, but also that striking economies are effected when compared with the cost of liquid cyaniding. The operating cost is claimed by the manufacturer to be only 40-50 per cent of that of salt bath treatment.

INSTITUTION OF MECHANICAL ENGINEERS

Forthcoming Meetings of the Automobile Division

The following meetings will be held during December:—

BIRMINGHAM CENTRE

Tuesday, 15th December, 6.45 p.m., in the James Watt Memorial Hall, York House, Great Charles Street. Paper: "The Application of Power Assistance to the Steering of Wheeled Vehicles," by F. H. Heacock, A.M.I.Mech.E., and H. Jeffery, A.M.I.Mech.E.

LUTON CENTRE

Wednesday, 16th December, 7.30 p.m., in the Town Hall Assembly Room, Luton. Paper: "The Manufacture and Properties of Automobile Suspension Springs," by C. J. Dadswell, Ph.D., B.Sc.(Eng.), M.I.Mech.E., J. E. Russell, M.A., and R. Fielding.

SCOTTISH CENTRE

Monday, 21st December, 7.30 p.m., in the Institution of Engineers and Shipbuilders, 39, Elmbank Crescent, Glasgow. Paper: "The Small High-Speed Two-

Stroke Petrol Engine," by J. C. Morrison, Ph.D., B.Sc., M.I.Mech.E.

WESTERN CENTRE

Thursday, 10th December, 6.45 p.m., in the South Wales Engineers' Institute, Cardiff. Paper: "The Jaguar Engine," by W. M. Heynes, M.I.Mech.E.

The following meetings will be held during January:—

LONDON

Tuesday, 12th January, 5.30 p.m., at The Institution of Mechanical Engineers, Storey's Gate, St. James's Park, S.W.1. Paper: "The Charging Processes of Internal Combustion Engines, with Special Reference to the Two-stroke Cycle," by Professor Dr. Hans List.

COVENTRY CENTRE

Tuesday, 5th January, 7.15 p.m. General Meeting in the Craven Arms Hotel, High Street. Paper: "Problems in the Design and Development of an

Economical Automobile Gearbox," by T. C. F. Stott, M.I.Mech.E.

DERBY CENTRE

Monday, 11th January, 7.15 p.m., in the Midland Hotel, Derby. Paper: "Continental Cars," by Laurence Pomeroy.

LUTON CENTRE

Monday, 11th January, 7.30 p.m. General Meeting in the Assembly Room, Luton Town Hall. Paper: "The Jaguar Engine," by W. M. Heynes, M.I.Mech.E.

NORTH-WESTERN CENTRE

Friday, 15th January, 7.15 p.m. General Meeting in Reynolds Hall, College of Technology, Manchester. Paper: "The Charging Processes of Internal Combustion Engines, with Special Reference to the Two-stroke Cycle," by Professor Dr. Hans List.

WESTERN CENTRE

Thursday, 7th January, 6.45 p.m. Informal Meeting.

FLUORESCENT TUBES

AN announcement has recently been made by The General Electric Co. Ltd., to the effect that their fluorescent tubes will henceforward have greater reliability and efficiency. In particular, the Osram 5 ft, 80 watt Daylight and Warm White tubes, which are the ones generally used in industry, will give an increased light output of 12½ per cent.

This advance has been made as a result of new techniques and improved manufacturing methods.

It is of interest to note that the luminous performance of the powders used to coat tubes of this type can be greatly modified by minute traces, in the order of a few parts in a million, of impurities. The chief contributions of

research to the improved tubes are the finding of the best formula for, and the best structure of, the fluorescent powder; the discovery of the best way to deposit the powder on the inner surface of the tube, and how to make it adhere permanently; and thirdly, the design of a cathode to produce the best life performance.

THE MIDLAND RED S.14

Advanced Features of Design Incorporated in a Single-decker Service Bus of Integral Construction

ONE of the largest passenger transport organizations, the Birmingham and Midland Motor Omnibus Co. Ltd. constructs its own vehicles. It is, therefore, in a position to lay down its own specifications in their entirety, and the post-war additions to its large Midland Red fleet have established a reputation for the progressive quality of their designs.

The latest product from the Carlisle Road works in Birmingham, the prototype for 270 single-decker service buses, is in many respects revolutionary in character. Known as type S.14, it is at present undergoing stringent testing at the M.I.R.A. proving ground. It embodies such modern features as integral construction of body and chassis, underfloor engine, automatic transmission, independent front-wheel suspension, bonded rubber-metal suspension units at front and rear, disc brakes, a disc-type transmission brake, and a hypoid final drive. It weighs less than 5 tons unladen.

Some of these features have already been well proved. When vehicle production was recommenced soon after the end of the war the underfloor engine was introduced. It has been fitted to no fewer than 700 single-deckers and has in the aggregate covered approximately 150 million miles. This 8-litre engine is retained in the new S.14 design for two reasons; (a) the reduction in vehicle weight will reduce engine stresses and allow maintenance overhaul periods to be extended, whereas this would not be the case were a smaller engine to be used, and (b) the use of an existing engine, that is, one already in production, naturally makes for economy in manufacturing costs. It is a six-cylinder, direct-injection diesel of 113 mm bore and 133.3 mm stroke, developing 100 b.h.p. at 1,750 r.p.m. and a maximum torque of 350 lb-ft at 1,100 r.p.m.

Another post-war introduction was the all-steel body, built to B.M.M.O. design and specification by the larger body-building concerns specializing in this form of construction. The high performance of these vehicles, both from the operating and maintenance points of view, led to the production of two prototypes of monocoque or integral chassis-body construction. The first of these prototypes was built entirely in steel and has now completed 200,000 miles. The second was entirely of light alloy, with a stressed skin, and has covered 70,000 miles.

It is following the experience thus gained that the S.14 has been designed. It is of integral type, with a stressed skin, and is virtually of all-steel con-

struction, the only light alloy parts being the internal panels or stressed skin, the exterior panelling and the doors. The aims of the designers were the elimination of dead weight for a given strength factor, greater fuel economy, lower maintenance costs and increased passenger comfort.

Independent front suspension, of conventional coil spring and superimposed wishbone type, and the bonded rubber-metal rear suspension units have also been used on the two previous prototypes, so that considerable experience of these items has now been obtained.

A vehicle equipped with the Hobbs transmission has also been in regular service for several months. In its latest form, as fitted to the prototype, the Hobbs box has automatic change for the second, third and fourth gears in both directions, up and down. A mechanical overriding device, which allows the driver to make a quicker selection of his gears in case of necessity, is also now provided in the form of a small kick-down pedal.

Integral construction

The S.14 is built to the 30 ft by 8 ft overall dimensions and is provided with seats for 44 passengers. The various stresses of operation are distributed through the stressed-skin body sides and the structural members which form the underframe. Both

stressed inner skin, the underframe, the roof, and the front and rear ends. Each of these is built on a jig, and final assembly is made on an assembly jig, the front end framing then being added.

The underframe is designed to receive the engine, gear box and suspension without the position of the body pillars and windows having to be considered. It consists of a series of cross members of deep lattice-girder type, the maximum depth varying up to 10 in, which accommodate the front and rear suspension assemblies, and a number of channel-section pressings which are spaced to suit the mounting of the power unit and other components. In building up these members 2 in angle material of $\frac{1}{4}$ in thickness is used. The part of the framing that carries the rear suspension resembles a rectangular box of deep lattice girders. The members forming the front and rear of this box extend the full width of the vehicle and fit within two oppositely positioned crib-rail angles which extend the full length of the vehicle and are part of the side structures. Other deep lattice members brace the box fore and aft, being splayed out to join up with the two crib-rail angles.

Also of deep lattice-girder form is the framing for the mounting of the independent front-wheel suspension.

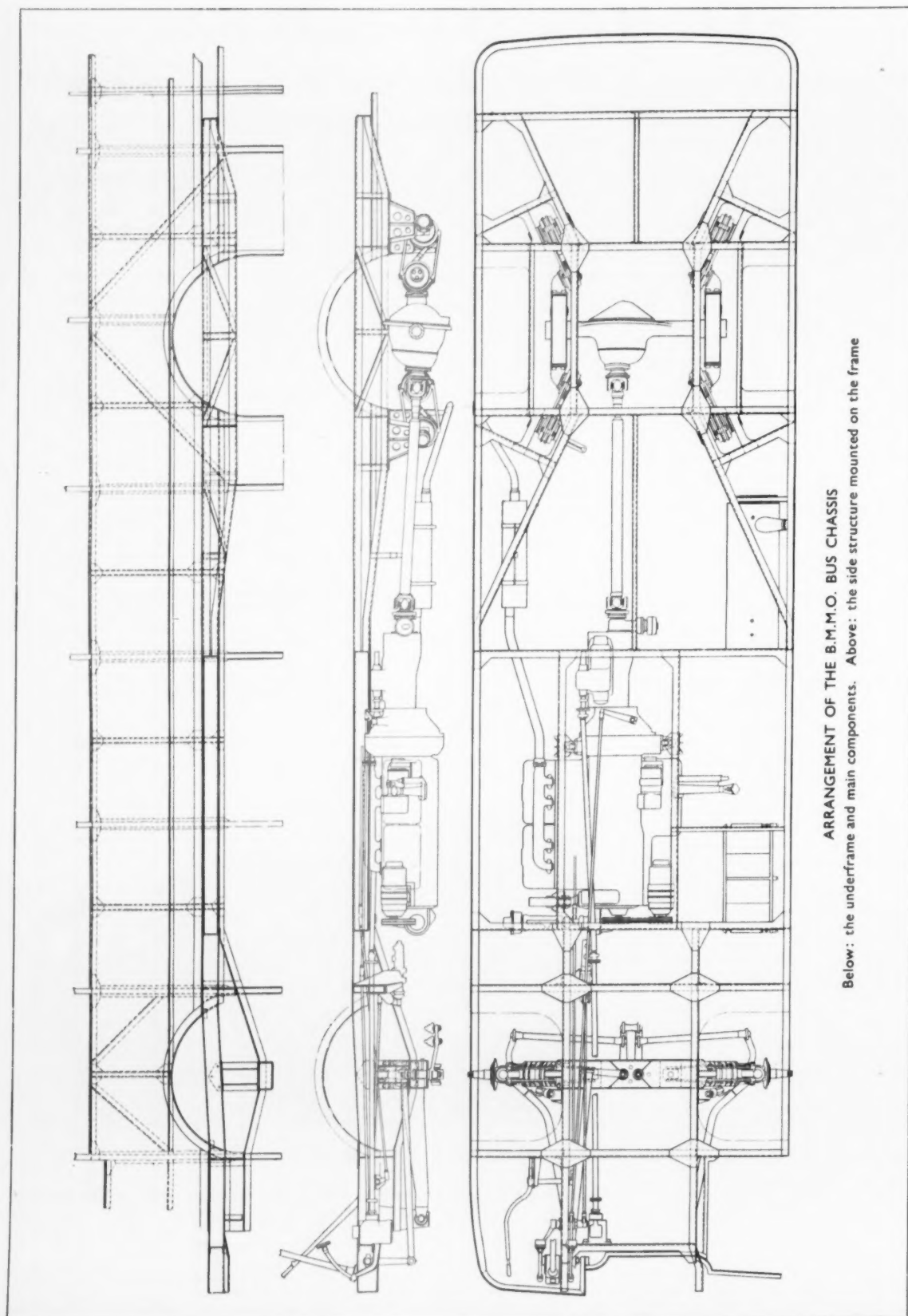


The Midland Red S.14 prototype

welding and riveting are employed in the construction, solid steel rivets being used in positions of stress while aluminium pop rivets are used to secure unstressed parts such as exterior panelling.

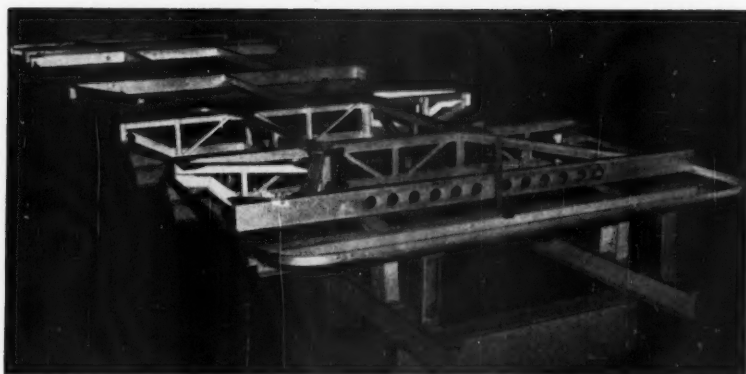
There are six main sub-structures, consisting of the two sides with their

It is reinforced with box-like brackets which support the suspension assemblies, and the lattice structures are bridged by a transverse member of box section. To the underside of this member are welded the brackets which form the mountings for the inner ends of the lower wishbones. At floor level



ARRANGEMENT OF THE B.M.O. BUS CHASSIS

Below: the underframe and main components. Above: the side structure mounted on the frame



The underframe structure on its assembly jig. View from the rear

the lattice structures are bridged by a member of inverted channel section.

The side framing is built up of pillars to which gusset-type brackets are welded to join up with the fence rail and cant rail. The jig for the assembly of the side framing is double-sided, the outer surfaces of the two sides as they take shape facing inwards towards one another at an angle.

The pillars are built up from 1½ in square tube and 1½ in top-hat section with ¾ in flanges, both of 18 gauge. At waist level the two sections are joined together by a short 18 gauge square tube dovetailed between the tube and top-hat sections so as to form a complete pillar, the halves being finally spot-welded together. This particular arrangement was adopted in order to secure maximum pillar strength at waist level with a maximum saving in weight, while the top-hat section facilitates the riveting of the stressed skin to the pillars.

For the fence rail, short lengths of inverted channel section are used, joining each pillar to its neighbours, but the cant rail is continuous and forms part of the roof structure. The two continuous crib-rail angle sections of 2 in material ½ in thick are riveted to the flanges of the top-hat section of the pillar and are spaced 4½ in apart so as to receive the extremities of the cross members of the underframe.

The light alloy stressed inner skin of 18 gauge is inserted between the inner faces of the pillars and waist-rail and the outer faces of the two crib-rail angles. Stump-type pillars of channel section are positioned centrally between each pair of main pillars and join the fence rail and the crib rails, to which they are welded and riveted, small gusset plates being used at the joints. The stump pillars form the joint line for the panels of the light alloy inner skin and also provide for the fitting of anti-drumming material for the outer panels, the joint lines of which are on the centres of the main pillars.

The lower crib-rail angle is continued over the wheel arch and is reinforced by two diagonal bracing members joining the stump pillar to the main pillar and to the wheel-arch

angle. The structure at this point is completed by the solid riveting of the 16-gauge light alloy wheel boxes to the wheel-arch angle and the underframe, and by solid riveting the main side-structure members with steel rivets and the stressed skin with light alloy solid rivets.

Top-hat section of 16 gauge is used for the cross members and longitudinal members of the roof. The cant rails are of 18-gauge channel section and, with the cross and longitudinal members, are built up on a jig prior to the final assembly, when they are solid riveted and welded to the gusset brackets on the tops of the main pillars. Finally, the exterior and interior lining panels are pop-riveted in place after the whole vehicle structure has been assembled.

There is no separate driver's cab, the driver using the front near-side entrance door. A full-depth partition is provided behind the driver's seat and is secured to the adjacent off-side pillar. On the central-gangway side the partition turns forward slightly and is joined to the front end structure by a shallow partition over which the driver steps to reach his seat. The entrance door is of gliding type and is operated electrically by the driver through switches controlling a C.A.V. electric door

motor with G. D. Peters' chain-drive gear.

Side windows are the Hallam, Sleight and Cheston "Famco" type with double sliders, each window assembly being encased in its pan with a continuous glazing rubber. Windows are in ¼ in glass and the front screens are of ⅜ in safety glass, that in front of the driver being raked in order to avoid reflections.

Seats are the B.M.M.O. light-weight type developed in conjunction with Accles & Pollock and weighing only 25 lb when trimmed. Heating is by a Clayton Dewandre C.H.V. recirculatory heater positioned beneath a seat at the mid-length of the vehicle. A Clayton Dewandre S.8 demister supplies four air-distribution slots in the front panelling. The 100 amp-hour batteries, master switch, control board and battery-boost unit are mounted on the rear side immediately to the rear of the entrance, a hinged flap giving ready access to them. The floor is of ½ in plywood, in two portions, with a butt joint on the vehicle centre line. It is supported by top-hat section members forming part of the underframe, has a P.V.C. covering of light weight, and floor treads in the gangway.

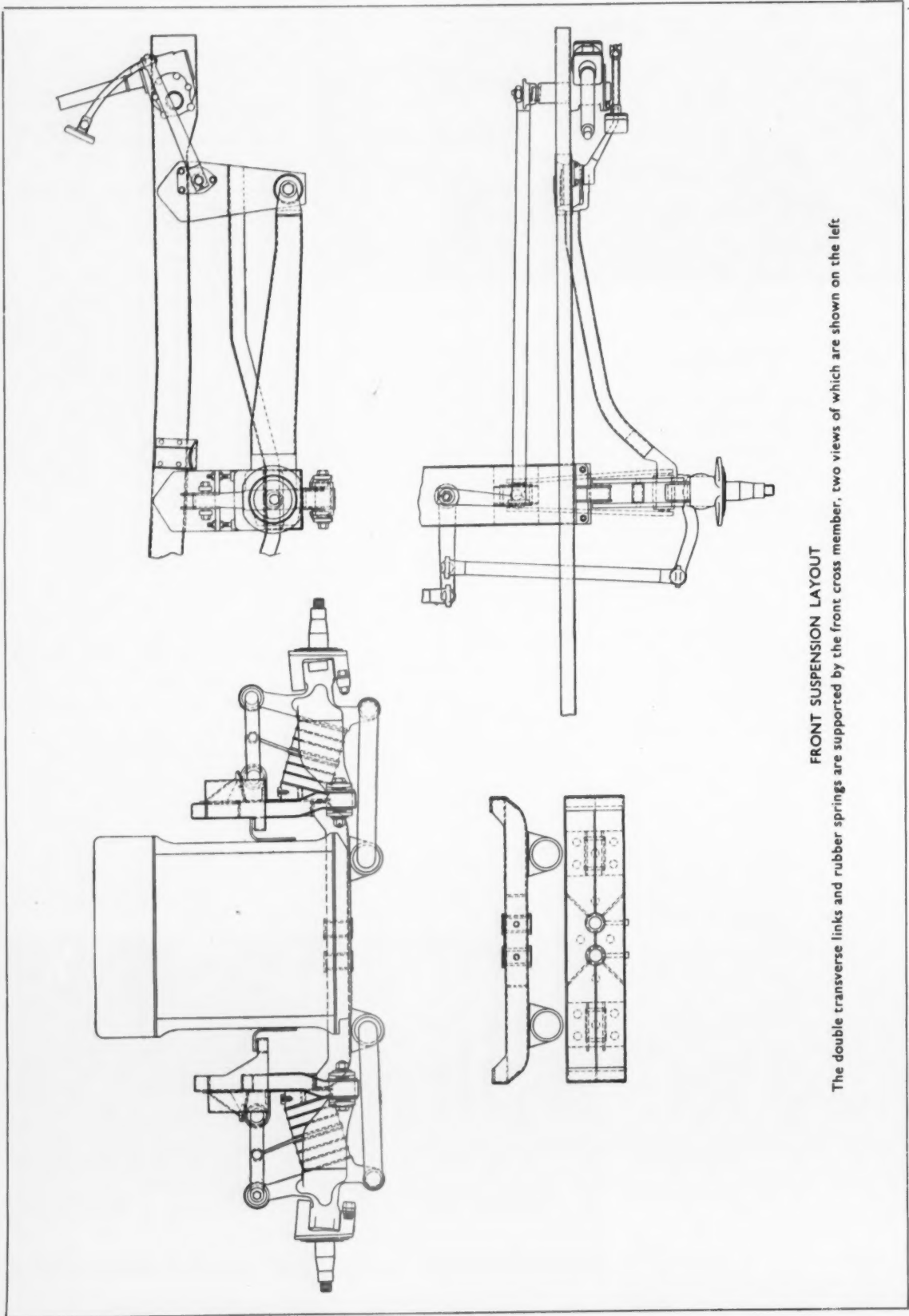
Front suspension

Front suspension is similar to the I.F.S. employing coil springs, as already used on a number of vehicles in service, except for the use of bonded rubber-metal suspension units developed in conjunction with Metalastik. The rubber is in shear and in compression, and the units are similar to those which have long been utilized in railway rolling stock.

Each unit consists of three layers of rubber bonded to metal plates, the end plates having bolt holes for their attachment. Two such units are attached to a centre plate, the upper end of which is hinged to the upper wishbone in order to resist any tendency of the spring to buckle. The other end plates are bolted to the front-wheel swivel standard and to a wedge-type bracket secured to the box-section bracket on



Both side structures in position on the underframe



FRONT SUSPENSION LAYOUT
The double transverse links and rubber springs are supported by the front cross member, two views of which are shown on the left

the underframe structure. Under load the spring is positioned with its axis at approximately 16 degrees to the horizontal. Provision is made for adjusting the height of the suspension in the event of any rubber creep, although it is anticipated that this will be negligible.

Superimposed wishbone links connect the wheel swivel standards to the underframe structure, and a box-section horizontal radius rod is bolted to the standard and extends forwards to a box-section bracket on the underframe; it thus locates the wheel and takes brake torque-reaction stresses. The length of the radius rod is considerable in comparison with the vertical displacement of the wheel, and in consequence the alteration in castor angle is small and has not proved to have any noticeable effect on the behaviour of the vehicle on the road.

Rubber bushes are used for the forward anchorage points of the radius rods and for the four pivot points of the wishbones, Metalastik ultra-duty bushes being used for the outer ends of the lower wishbones and spherical-type bushes at the other points.

From the foregoing description of the layout of the independent front suspension, the general manner in which the bonded rubber spring is loaded will be clear. Part of the weight is supported by the vertical component of the shear force in the rubber and part by the vertical component of the compression force. If the top and bottom transverse links are horizontal, as they are intended to be under the normal laden weight, then the vertical reaction at the hub is as follows:—

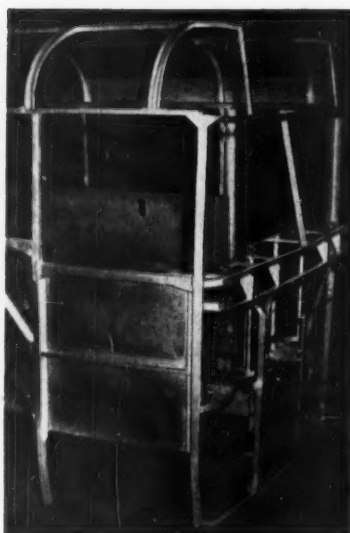
$$P = S \cos^2 a + C \sin^2 a$$

where P is the vertical reaction, S is the shear force in the rubber, C is the compression force in the rubber and " a " is the angle of the bonded surfaces to the vertical.

In the actual case of the Midland Red bus the angle " a " has a tangent of 0.3, so that the formula can be written

$$P = \frac{1}{1.09} S + \frac{0.09}{1.09} C$$

but no fixed rule can be given for the



The front-end structure is a separate sub-assembly

optimum angle or for the correct ratio of S and C . The design was worked out so that the ratio of the compression deflection to the shear deflection of the rubber is 0.3, that is, $\tan a$, but a different ratio of deflections could have been used. The compression and shear do not have to be removed in equal proportions when the abutment of the spring on the chassis is moved vertically upwards.

The non-linearity is due only in a very small measure to the stiffness characteristic of the spring, which for practical purposes can be assumed to obey Hooke's law. It is due almost entirely to the swinging of the transverse links. The effect is the same whether the long trailing arm is used or a corresponding leading arm or a double wishbone construction. The load supported in the attitude in which the links are horizontal is the same as if the axle were constrained to move vertically, but the stiffness is not the same. The reason is that if there is

any displacement of the spring in either direction the links become inclined to the horizontal, and the tension in them has a component tending to restore them to the horizontal position. This is analogous to a pendulum effect in which the horizontal component of the force in the rubber replaces the gravitational force on the bob of the pendulum.

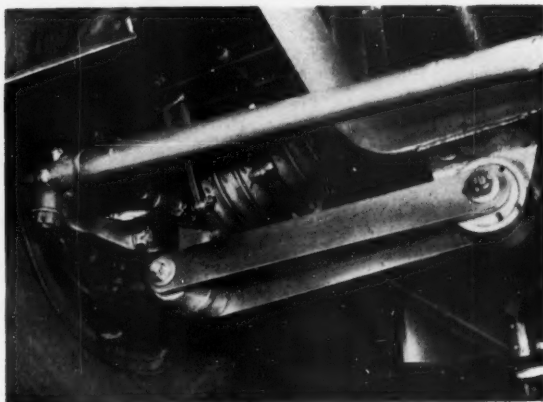
Any doubt about the added stiffness due to this cause can be removed by considering how the suspension would work if the bonded plates were absolutely vertical. In that case, if the links were horizontal, the pre-compression would not help in the load capacity but would add to the stiffness. In addition to this fundamental principle, the following detail points are of some importance:

1. The stiffening which results from the swinging of the links may appear at first sight to be a disadvantage because there is a higher periodicity for the same storage of strain energy in the spring material. The same argument can be used, of course, against most forms of non-linear spring and, indeed, the increased weight is one of the reasons why constant periodicity has achieved little popularity with users of steel springs.

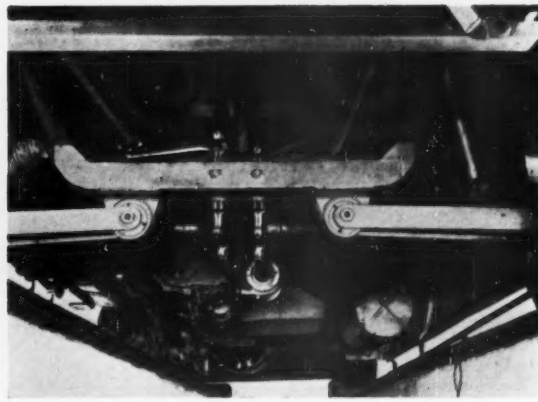
2. In the case of the rubber spring the disadvantage is not a real one. A certain amount of strain energy both in shear and in compression remains in the spring at zero supported rate, but it is a condition for maximum strain range in rubber that the value of the strain should keep all the time definitely on the same side of zero.

3. In the toggle link rear springing, the toggle effect does not increase the stiffness when the links are in line. This is because the bushes are concentric and the resistance is due entirely to torsional shear of the rubber. If the bushes were eccentric with the linkage in its aligned position, there would be a change in stiffness.

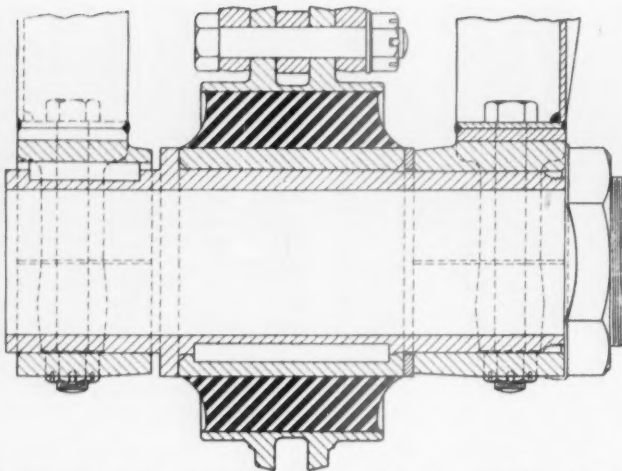
4. With the proportions of the top and bottom links actually used, the curve is a compromise between the straight line and the logarithmic constant periodicity. To achieve the latter,



Arrangement of wishbone links, rubber spring unit, and box-section radius rod. Underneath view from rear



Mounting of lower wishbones and the divided track rod of the steering layout. Underneath view from front



Diagrammatic section through a rear suspension unit mounted on the frame

a considerably shorter bottom link would be needed and the periodicity would be well above its present value, either light or at full load.

5. The compression stiffness of the spring is $17\frac{1}{2}$ times its shear stiffness and this ratio would be increased, of course, if there were more intermediate metal plates.

6. Due to the direction of the resultant force in the rubber spring there is a large continuous tension in the lower transverse link and normally very little force in the top link. It is considered a definite advantage with the difficult lower wishbone links to have the load always in the same direction, not reversing continually as it does with normal springing. As a tension member the bottom link can also be constructed fairly lightly.

7. The spring is described, of course, as loaded in compression and in shear, but in practice it is impossible to get shear without bending. The intermediate steady plate which connects to the middle of the top transverse link does not affect in any way the shear and compression movements but keeps the centre cross-section of the spring parallel to its built-in ends, thus eliminating the greater part of the

bending effect. The initial off-set of the springs is also aimed at reducing the bending, and it is under consideration to increase the off-set in a new or modified design by stepping the inner element of the spring upwards by $\frac{1}{4}$ in to 1 in relative to the outer. This can be done in the attachment to the intermediate plate, which can be made with dowels in place of bolts. The compression is so much greater than the shear that there is no danger of slipping or fretting.

8. If the off-set were chosen accurately there could be no bending moment on the steady plate in the mean position. It does not follow, however, that the steady plate could be omitted. If it were, the central plates would still rock through a large angle as the spring moved up and down.

9. The different characteristics for the very light bus and for the luxury coach to which this suspension is also being fitted are achieved by change in the rubber mix, but the difference in weights is not so large on the front axle as on the rear axle. Eventually it should be possible to reduce quite considerably the size of the rubber springs for lightweight buses.

10. Although the internal damping of the rubber is not sufficient to enable the bus to run without shock absorbers, it does seem that the size of the dampers can be reduced from that found necessary in conjunction with coil springs.

Rear suspension

For the rear suspension Metalastik units employing rubber in torsional shear are used, following their successful behaviour over nearly a quarter of a million vehicle miles. Four such units are employed at each end of the rear axle, two being in front of the axle and two behind it. The units of each pair are connected together by two toggle links joining their outer metal casings or sleeves, their inner metal bushes being secured respectively to the mountings on the underframe and on the axle casing. The pairs of units

are also set at an angle to the longitudinal axis of the vehicle, so that seen in plan they form a wide-angle vee. Accordingly they position the axle and tend to prevent rolling on corners.

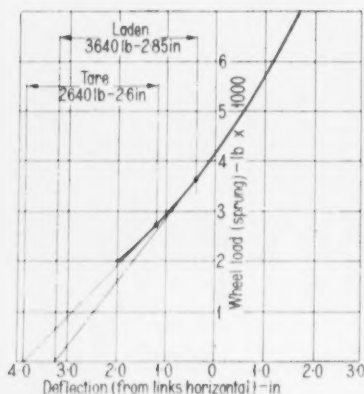
The four rubber units at each side lie with their centres in the same horizontal plane when the vehicle is unladen, and the load is carried entirely by these bushes with the rubber in torsional shear. With deflection there is, however, eccentricity of the inner sleeve relative to the outer sleeve, and the resistance introduced in the rubber by this eccentricity is in addition to that due to change in torsional deflection. The eccentricity causes increased stiffness as the deflection increases, and over the normal load-deflection range from unladen to laden the stiffness of the system is proportional to the load carried. The result is that the ride is uniform whether the vehicle is empty or full.

Further spring travel due to overload or bump increases the stiffness rate more rapidly, while there is also progressive stiffening of the springing for the rebound position. The outer sleeve of each bush is provided with jaw-shaped extensions which accommodate the toggle links, and there are in addition diagonally placed turn-buckles which provide for adjustment and for the re-setting of the suspension due to any slight settling which may occur in the rubber after a period of service. Newton Bennett direct-acting hydraulic dampers control both front and rear suspension units.

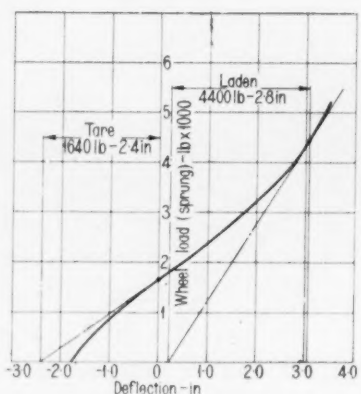
Disc brakes

The Girling disc brakes on the front wheels have six hydraulic cylinders and pistons applying the brake pads direct, three at each side of the disc, these being accommodated in a caliper member which is arranged with its mid-point approximately on the neutral axis of the wheel, so that the action of the brake shall be unaffected by any rocking of the wheel on its bearings due to necessary clearances or, in the course of time, to wear.

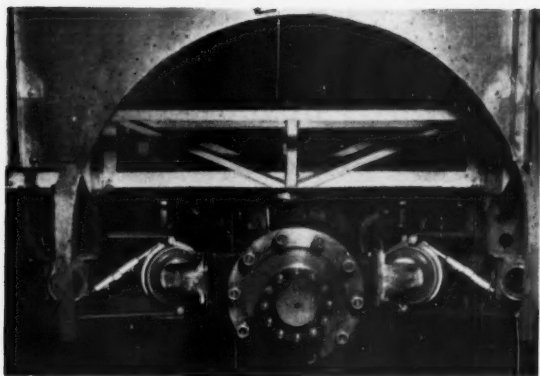
The rear disc brakes have four hydraulic cylinders to each side of the disc, and they operate the friction pads



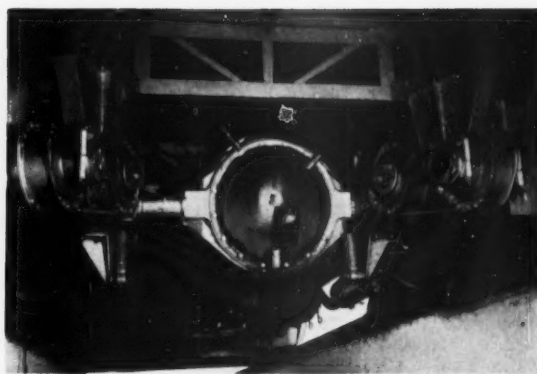
Load-deflection curve, front suspension



Load-deflection curve, rear suspension



Arrangement of rubber-metal rear suspension units, toggle links and turnbuckle-type adjusters



Mounting of the rear suspension units. View below the vehicle, looking forward

through rocking levers so that the hydraulic system is thus removed from the source of heat. The system incorporates the continuous-flow hydraulic servo, produced by Automotive Products, which is used on all the vehicles of the B.M.M.O. single-decker fleet.

Fitted to the transmission as an experiment is a manually-operated disc brake, for the designers consider that only on the transmission will a hand-brake provide the efficiency given by the footbrake. It is, therefore, desired to obtain data and experience for later reference in the event of transmission brakes being legalized in the future. The disc is mounted on the pinion shaft of the final drive and it has a single friction pad at each side of the disc operated through rocking levers and a Girling wedge-type expander.

Regarding the power unit, the engine in the prototype has a light alloy crankcase with a separate cast iron cylinder block, but this will be superseded by a combined crankcase and cylinder block in cast iron. Pistons are of toroidal-cavity type and are produced in silicon alloy. The big-end bearings are of lead-bronze, and the seven-bearing crankshaft is Tocco hardened and is carried in half-and-

half lead-bronze and white-metal bearings with steel main bearing caps. The cast camshaft is rigidly supported in seven bearings and has hardened bearing surfaces and cam profiles. It is chain driven from the crankshaft and a manually adjustable chain tensioner is provided.

Lubrication is arranged on the wet-sump system, with the oil pump mounted on the front main bearing cap and driven from a gear at the front end of the crankshaft. The oil sump is a one-piece casting with its rear half arranged as an oil reservoir with a capacity of 4½ gallons. Oil is drawn from the sump through a wire-gauze filter and passes through a close knife-edge type pressure filter to the main bearings by way of an oil gallery mounted on the outer faces of the bearing caps.

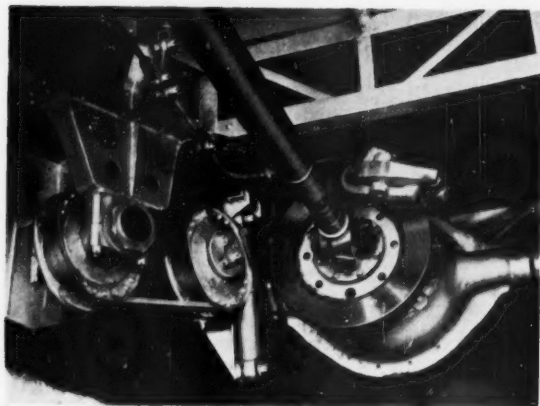
The fuel injection pump drive is through a small bevel gear box mounted in the front end of the engine, the drive being taken off an extension of the camshaft. The pump itself lies parallel to the axial centre line of the cylinders and points towards the off-side of the vehicle. The power unit is supported on two Metalastik conical rubber bushes at the rear end and by a single parallel-type bush at the front end.

Either the standard design of B.M.M.O. four-speed gearbox or the Hobbs automatic transmission can be employed and in both cases they are arranged for unit construction with the engine; the gearbox casings are of light alloy. If the standard gearbox is utilized a single dry-plate clutch is used.

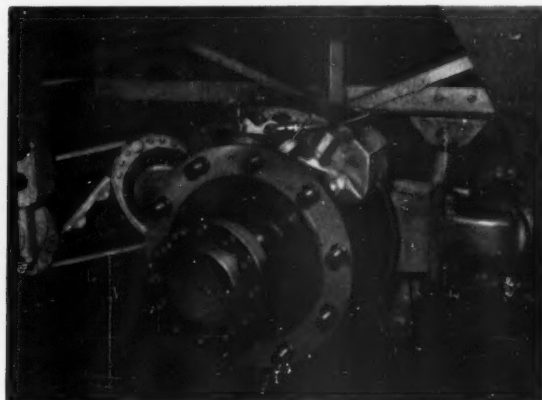
A hypoid bevel final drive is fitted, following successful experimental operation over some thousands of miles. The open propeller shaft is equipped with Hardy Spicer universal joints.

Steering is by a Marles double-cam unit and in order to obtain minimum turning circles to off-side and near-side the track-rod is divided. The inner ends of the two links are supported by swing levers mounted on vertical shafts carried in a bracket off the box-section cross member of the suspension. The upper end of the off-side vertical shaft carries a steering lever which is attached to the rear end of the drag link. The steering unit is arranged with the drop arm facing inwards towards the centre line of the vehicle.

Both front and rear wheels carry single 9.00 × 20 in tyres, following satisfactory service by single rear tyres on a number of Midland Red vehicles for a considerable period of time.



A disc brake on the pinion shaft is manually operated through a Girling wedge-type expander and rocking levers



The rear wheel disc brakes are hydraulically operated but the friction pads are actuated through rocking levers

A COMMON TOLERANCE SYSTEM

The "A.B.C. Proposals" Critically Examined

F. W. M. Lee, M.I.P.E.

AMERICA, Britain and Canada (A.B.C.) are endeavouring to establish a Common Tolerance System, known as the "A.B.C. Proposals," to ensure interchangeability of manufacture. For this purpose the British Standards Institution (B.S.I.) has submitted for consideration a conversion of the Continental I.S.A. System from metric to the inch equivalents. Such a proposal must satisfy certain essential requirements and in reviewing the problem it appears that there are four broad aspects which should be considered.

- (1) What is the fundamental basis of the I.S.A. System?
- (2) To what degree does the theoretical conception of the I.S.A. System satisfy practical requirements?
- (3) Will the conversion from millimetres to inches contribute to or detract from the I.S.A. System?
- (4) Is the conversion suitable for use as a Common Tolerance System?

It is not possible in the course of an article of limited length to give each of these questions the comprehensive investigation it merits, and the following remarks are confined to the salient features only.

Fundamental basis of the I.S.A. System

Reference to the B.S.I. publication B.S.164, Part 1: 1951, and to other sources indicates that the I.S.A. System is derived from:

- (1) Preferred numbers.
- (2) Existing Continental practice.
- (3) A geometric series.
- (4) An arithmetical progression arranged in a purely arbitrary fashion.
- (5) The following formula: "The 'fundamental tolerances' are multiples of the fundamental tolerance unit 'i' where $i = (0.001 \text{ in}) = 0.052 \times \sqrt[3]{D} + 0.001 \times D$ (D in inches)."

To the practical engineer the following might be more acceptable:

- (1) Preferred tolerances.
- (2) Based upon modern British and American practice.
- (3) In easily understood steps.
- (4) A formula in keeping with the mathematical capacity of the majority who will be using it.

By definition, a "system" is "a regular method or order" which cannot apply to the I.S.A. since the toler-

ance values have been collected from a number of unconnected sources and assembled as a table.

Dominating feature of the I.S.A. System

The tolerance steps from H5 to H6 to H7 and onwards, as plotted in Fig. 1, show that the aforementioned cube root formula sets the general pattern from which it will be noted that the curve rises slowly at the commencement, increasing its slope reasonably for a short distance, then running away at an extremely steep angle to the end. The significance of this curve and the tolerance steps it gives will be referred to in the remarks concerning the "I.S.A. in Use."

Tolerances not calculated

It is evident from an inspection of Fig. 2, constructed upon the tolerance

the qualifying statement "the inch tolerance values have been obtained not from the I.S.A. tolerance formulae, but by direct conversion of the metric values."

It follows that a tolerance for any diameter not given in the tables could not be calculated and would, therefore, have to be decided by protracted international agreement, which is impractical.

Remaining features of the theoretical I.S.A. System

- (1) There are sixteen grades of manufacturing accuracy.
- (2) Twenty-one types of fit.
- (3) Particulars given for the manufacturing tolerance on the gauge ends themselves.
- (4) The amount the gauge may wear before withdrawal from service is stated.

- (5) Recommendations for fits are provided with examples.

- (6) A claim is made that if the grade of the hole and the shaft are confined to the same qualities of manufacture, as, for instance, H7 g6, the "feel" will be the same irrespective of the diameter.

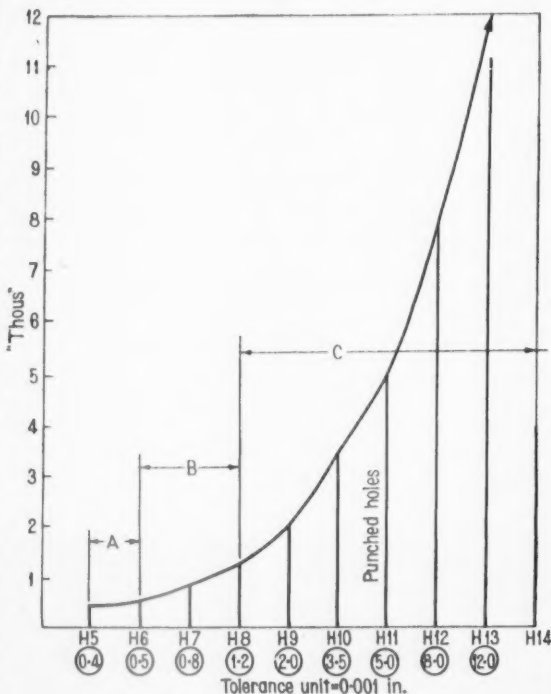
The I.S.A. in use

It will be realized that the I.S.A. System, whilst partly based upon existing practice, is essentially a theoretical conception, and it is proposed to show how and why the original has been extensively modified in practice.

From B.S.1916, Part 1: 1953, pages 16 to 23, I.S.A. conversion, it will be noted that there are limits for holes from oversize to unilateral to bilateral to undersize consisting of 64 possible tolerances for a 1 in. diameter hole. In France and Holland, however, this has been reduced to the unilateral series H6, 7, 8 and 11. In this connection it can be said that the three tolerances H6, H7 and H8 are suitable for fits and H11 for clearance holes.

Similarly, from pages 24 to 31, there are 74 possible tolerances for an inch shaft as compared with the actual practical application by I.S.A. users of 7 degrees of precision characterized by the numbers 5, 6, 7, 8, 9, 10 and 11. The range, however, may be increased to cover exceptional conditions of service.

From the broadsheets issued by



In the unilateral system the smallest diameter of any hole is nominal and therefore the ringed figures are the number of "thous" the largest diameter of the hole is greater than the nominal size, as represented by the "not go" end of the gauge.

A=Rise too slow: B=Rise reasonable: C=Rise too steep

Fig. 1. Tolerance values for 1 in diameter hole, unilateral "H" series B.S.1916, Part 1: 1953

proposals given in B.S.1916, Part 1: 1953, that the tolerances were not calculated, since calculation would ensure that the curves would not collide at points A and B. That this was realized during the conversion from millimetres to inches is confirmed by

engineering firms in France and Holland it will be found that the unilateral system only is specified. It will, therefore, be realized that the original theoretical conception of the I.S.A. System has been drastically reduced in practice to virtually a three-hole system (Newall A, B and C). Investigation of nearly 10,000 limits has already shown this to be inadequate and will not satisfy the requirements of the whole of the engineering industry with the inevitable result that the existing difficulties of "special tolerances" will be perpetuated.

There must be a sound practical reason for this limitation of scope to arise and further investigation discloses it. As already mentioned, the fundamental basis of the I.S.A. System is a cube root law producing a tolerance step curve as shown in Fig. 1.

Inspection of this curve shows the slope to be so slow at the commencement that the maximum diameter of a 1 in. H6 hole is only one tenth of a "thou" larger than an H5 hole, which for practical purposes is useless and has been discarded. The slope then increases and provides suitable steps for the following three tolerance grades of H6, H7, H8, that is, 5 tenths, 8 tenths and 12 tenths tolerance for a 1 in diameter hole.

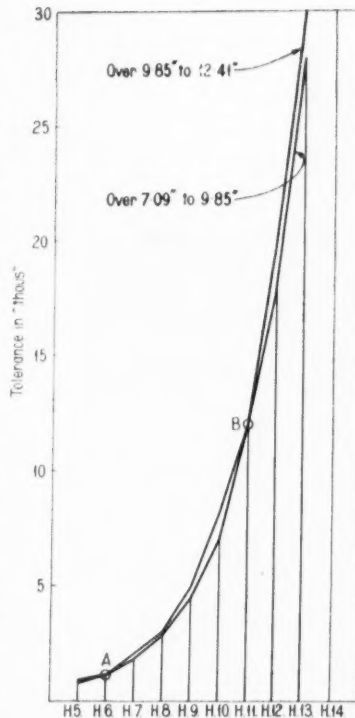
Subsequently the curve commences to rise more rapidly and the tolerances for H9 and H10 become too large to be considered suitable for customary fits. H11 is used as a rough check on drilled or punched holes. From H11 onwards to H16 the slope is so steep that the tolerances provided are meaningless, since the majority from H13 onwards are so large that they can be measured with a steel rule.

It will be clear that the fit tolerances of H6, H7, H8 have only been chosen because they happen to fall upon that part of the curve the slope of which is suitable for the provision of practical working tolerance steps and the remainder, which form the greatest percentage of the original conception, have been ignored. A tolerance system, if it is to be thoroughly comprehensive, must provide tolerances suitable for watches to warships, and the limited range of H6, H7 and H8 could not possibly be considered adequate to embrace this very wide field.

Having in mind the extremely complicated nature of the I.S.A. System (see B.S.1916, Part 1:1953, page 54, Appendix B and C) the question was asked on the Continent if it was found to be complicated in use. The answer was "No, after the general idea has been grasped." Since the I.S.A. System has, by practical application, resolved itself into a simple unilateral, three-hole system of limited coverage and use the reply is understandable.

Gauge manufacturing tolerances

In Britain, the gauge manufacturing tolerance is 10 per cent of the limit, but in the I.S.A. System this is increased to 33.3 per cent, which no doubt will please the gauge maker, but not the user of the gauge, for it will



Collisions at A and B could not occur if tolerances had been calculated by the cube root formula.

Fig. 2. Tolerance step curves

be found that when the "Go" end has been made to its largest permissible diameter and the "Not Go" to the smallest permissible diameter, the gauging system has robbed the shops of no less than half of the drawing tolerance. This is not only unreasonable, but also unrealistic.

Permissible gauge wear before withdrawal

The I.S.A. System specifies this amount in tables supplied, but whereas the British practice is 10 per cent of the limit, the I.S.A. System varies, and in one example it is 14 per cent and another 21 per cent. These percentages are excessive and inconsistent.

Summarizing the I.S.A. System in use it has been found that owing to the cube root formula providing unsuitable tolerance steps:

- (1) Practice has virtually reduced it to a unilateral, three-hole system with limited coverage.
- (2) Gauge manufacturing tolerances can rob the shops of 50 per cent of the drawing tolerance.
- (3) The amount the gauge is allowed to wear undersize is excessive by British standards.

B.S.I. Conversion—B.S.1916, Part 1: 1953

Perusal of this publication shows that tolerances are provided for oversize, unilateral, bilateral and undersize holes with the result that there are 64 tolerances available for a 1 in diameter hole. Shafts are also amply provided for, since in many diameters 74 tolerances are given.

Bearing in mind the practical operation of the three-hole I.S.A. System the reason for the foregoing is difficult to understand. No doubt the number of tables could be reduced, for it will be noted that overlapping occurs, as for instance A9 and H9. A9 tolerances are +10 "thou" and +11 "thou" for a 0.040 in diameter hole. The smallest size of the hole is therefore 0.050 in, an increase of size of 25 per cent. The same hole could be specified by a 0.050 in hole in H9 which would be 0.050 in +1.0/-0.0 "thou."

Conversion affects interchangeability

The difficulty of converting millimetres to inches is well known because the inch fractional sizes obtained are unwieldy and become more complicated in the process of correlating them to an existing tolerance table. In the I.S.A. System the diametrical change steps are simple, 1, 3, 6, 10 mm for instance, but when converted become 0.040, 0.119, 0.237, 0.394 in and rounding is resorted to giving 0.040, 0.120, 0.240, 0.400 in. Odd unfamiliar sizes do not simplify matters.

Similarly, the tolerance values themselves have also been rounded with the result that in one case the difference between the actual I.S.A. tolerance and the inch conversion is nearly "half a thou" (4.8 tenths). Again, through rounding of the diametrical change steps it will be found that an inch-dimensioned component would be made to a different set of tolerance values than its millimetre counterpart, because one would fall into one range of tolerances and the other into another.

Further, a British or American designer would not consider making a shaft 0.3937 in diameter (10 mm) where he would normally employ one of 0.375 in ($\frac{3}{8}$ in) diameter, since the decision has already been made by the availability of raw material sizes.

The possibility of interchangeability is still more remote when the following are taken into consideration:

- (1) Different units of length.
- (2) Different tolerances due to rounding.
- (3) Different raw material sizes.
- (4) Metric threads.
- (5) Metric collets, feed fingers, etc.

While the ideal of interchangeability between millimetres and inches is highly desirable, its practical attainment is impossible.

Summarizing briefly the foregoing and making reference to further points which require serious consideration it will be found that:

- (1) The cube root law of the I.S.A. has little connection with practical requirements.
- (2) The practical application of the theoretical I.S.A. System shows that the greatest proportion of it is obsolete.
- (3) Owing to the unsuitability of the tolerances the I.S.A. can be said to have resolved itself into a unilateral, three-hole system.
- (4) There are insufficient suitable tolerances to cover the whole of engineering requirements.

- (5) The B.S.I. conversion perpetuates tolerance values which have already been proved to be of no value in practice and are redundant.
- (6) A perusal of B.S.1916, Part 1: 1953, Appendix "B" and "C" shows it to be complicated, a fact which has been commented upon in all three A.B.C. countries.
- (7) The diametrical tolerance change steps in millimetres are easily assimilated, but when converted to odd unfamiliar fractions of an inch become unwieldy.
- (8) Having considered the fit, tables must be searched in order to find the appropriate hole and shaft with the desired tolerances.
- (9) The manufacturing tolerance for the gauge can rob the shops of 50 per cent of the drawing tolerance.
- (10) The wear allowance on gauges before withdrawal from service is excessive.
- (11) Hope of interchangeability with the Continent has been destroyed.
- (12) Any tolerance not shown cannot be calculated and settlement must rest upon protracted international agreement.
- (13) By providing tolerances for both unilateral and bilateral systems the number of tables is doubled, gauge stocks enlarged and complication increased, for which there is no sound practical justification when establishing a new tolerance system.
- (14) The contention that the I.S.A. System provides the same "feel" for any fit when the hole and shaft are made to the same qualities irrespective of diameter and the proportion of diameter to length cannot be sustained until the human element has been eliminated and "feel" determined and measured mechanically.
- (15) No tolerances are provided for horological or instrument trades.
- (16) The tolerances for undersize holes are mixed up with the table for bilateral holes. See K6, K7, K8, M6, M7, M8, B.S. 1916, Part 1: 1953, page 20.
- (17) The I.S.A. System is completely divorced from existing British and American practice. By its adoption an enormous quantity of existing gauging equipment would become obsolete. The cost of its introduction would be fantastic.

It is the Author's opinion that a comprehensive and practical limit system should be so simple that it will neither tax nor defeat the ability of the draughtsman, the inspector, or the man in the shop and should be the servant and not the master of the user.

NEW THRUST BEARING

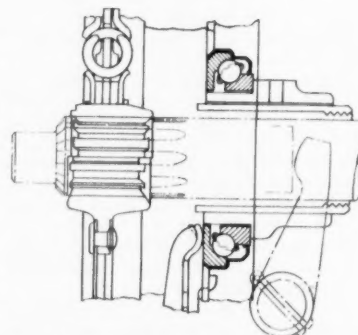
A Ball-type Bearing Designed for Clutch Withdrawal Mechanisms

CLUTCHES in modern motor vehicles have to operate under more and more exacting conditions as a result of increasing traffic congestion at home, and because of specific requirements of overseas markets. In consequence, the carbon block type of thrust bearing, frequently used in clutch withdrawal mechanisms, has sometimes been found to give an unduly limited life. The flat thrust-washer type ball bearing that is often used for this application is not entirely satisfactory, and trouble has been experienced as a result of ball spin. This spin takes place because the thrust is not great enough to prevent the balls from being flung outwards by centrifugal force and, as a result, true rotation of the balls on one axis cannot take place. Moreover, the grease tends to be flung clear of the cage and balls, and this condition is aggravated by the high temperatures caused by ball spin.

Ball bearings of the deep-groove type have also been used in clutches, but in some quarters they are not regarded as suitable for this application. The two-piece, ribbon type, pressed

cage employed in these bearings allows neither for slight inaccuracies in assembly in the clutch mechanism nor for any deviation from true thrust loading. Therefore, the track diameter of the path followed by the balls varies; and the acceleration and deceleration, caused by ball precession round the tracks, apply to the cage severe loads which sometimes cause it to fail.

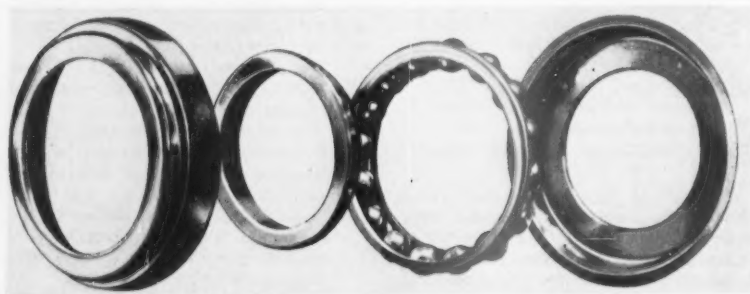
These considerations led the technical department of Pollard Bearings Ltd. to the conclusion that a bearing should be designed specifically for application to clutches. They decided to employ the angular-contact type of bearing, which is to a certain extent self-aligning. The self-aligning characteristic of this type of bearing arises partly because of its race form and partly because a large number of balls of relatively large diameter may be fitted within the limiting overall dimensions. Furthermore, a one-piece cage may be used, and this allows the balls a certain amount of circumferential freedom of motion. This freedom tends to prevent unduly high loading of the cage.



The new bearing in a Borg and Beck clutch

Developments have proceeded on these lines and a new range of clutch withdrawal bearings has now been introduced. These bearings are suited to pure thrust loading, and may be applied in conventional clutches as well as in automatically operated units. As can be seen from the illustration, the outer race has an end face of relatively large radial width and this forms an adequate bearing surface against which the clutch-operating levers may bear. The track form of both the inner and outer races is so designed that after the unit has been assembled, the races do not separate. Therefore, when the thrust load is released, the balls remain within the track, and operation is noiseless.

Pollard Bearings Ltd. state that tests so far carried out by independent motor manufacturers have shown that the life of this unit is more than fifteen times that of a standard clutch withdrawal bearing. So far, a failure has not been produced by these tests, after completion of which the bearings, when stripped, have been found to be well lubricated and fit for a much longer period of service.



A pressed steel shroud ring encloses the bearing assembly and not only retains the grease but also protects it from foreign matter

DEEP-HOLE BORING

Development of a New Production Process for High-speed Boring: Current German and Swedish Practice

H. J. Pearson*

DEEP-HOLE drilling is generally regarded as a difficult process to control on a production basis since it is not possible to observe machining conditions at the point of cutting. This difficulty of producing a deep hole—irrespective of whether it is necessary to obtain a high degree of surface finish—has resulted in component-design staff avoiding a potential bottleneck with work of this nature by fabricating the part necessitating a long bore from a number of small details.

On the Continent, the difficulties associated with this class of machining have, within certain limits, been overcome. Instead of being an operation difficult to apply in the sense that the unit production time is an uneconomic figure, it is now quicker to produce a deep-bored component than to process a number of shorter parts. To-day it is possible to drill a hole in bar material from 6 ft to 15 ft in length at a higher rate of production than that obtainable by conventional methods of drilling in a lathe or a horizontal boring machine.

A further and sometimes even more important consideration is that, when necessary, it is possible to eliminate secondary finishing operations as, for example, honing. Surface finishes of the order of 0.5 to 1.0 micro-inch r.m.s. at a penetration rate of 8-10 in/min have been obtained in holes as small as $\frac{1}{4}$ in diameter in lengths up to 6 ft.

These very high feed-rates are the result of developments from German gun-boring practice obtaining in 1945. The methods now used in Germany and Sweden are parallel with the difference in speed between drilling with a twist drill and the trepan-boring method used for heavy engineering components. Because of the specialized

technique entailed, existing practice in this country should not be used as a basis in assessing the remarkable results achieved when boring materials ranging from steels having less than 0.2 per cent carbon in their composition to stainless grades and light-alloys with a high silicon content.

Most of the materials used in the automobile industry are within this range, while the non-ferrous group extends to the brasses. In view of this, components such as engine gear-blanks, bearing inserts of all kinds, gudgeon-pins, valve-seat inserts, diesel injector and pump parts, hydraulic cylinders, and to a certain extent, cylinder liners could be produced in 6-15 ft lengths of bar-stock material and cut to length after boring at a far higher rate of production than by methods in use at the present time.

There are three principal methods of boring by the German technique, which makes use of tungsten-carbide tipped tools. Each of these requires the use of a tubular boring bar for the return of the swarf during machining as in existing trepan-boring practice. By this process, oil at high pressure is pumped around the outside of the boring bar to the point of cutting, and returns through the centre of the bar, carrying with it the swarf produced.

As shown in Fig. 1, the equipment consists essentially of a horizontal boring machine of fundamentally heavy-duty lathe design. A high-pressure oil head, such as illustrated schematically in Fig. 2, is brought into contact with the free end of the workpiece to provide an oil seal. In operation, the boring bar passes through the high-pressure head and is controlled by the longitudinal traverse of the boring-head carriage. During the machining, either the workpiece or boring tool can

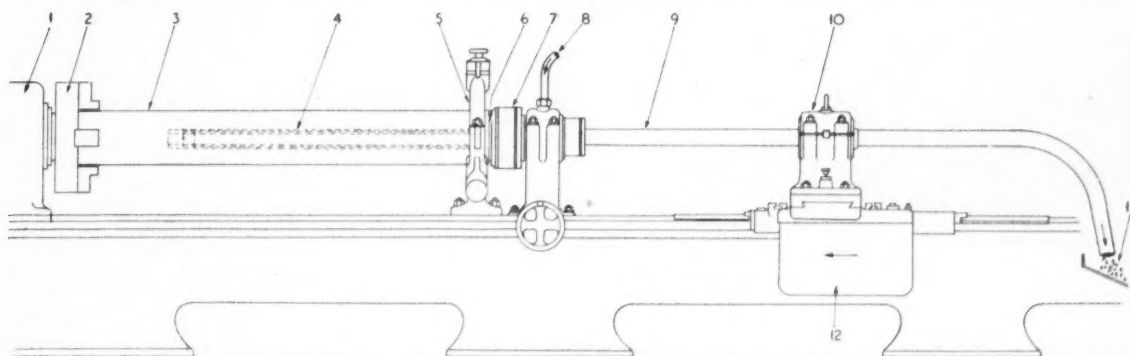
rotate, depending upon whether the workpiece is suitable for rotation, or both units can be rotated in a like or counter direction.

A pressure-head usually consists of two primary components: a stationary body, clamped directly on to the bed-ways of the machine or to a special saddle, and a free-running, circular-section member having either a rubber sealing ring or an entirely metal-to-metal seal. The bore through the body of the head and the rotating member is a slide fit on the boring diameter of the cutter. At the rear end of the head is a support bearing for the boring bar. Usually these bearings are in the form of an externally tapered, split-section hardwood bush to assist in damping-out vibration. With reference to Fig. 2, it will be seen that as the oil enters the head it is prevented from escaping in any way other than through the boring bar. As shown in other examples of actual heads, the rotating member is usually carried on one or more sets of ball-bearings.

Factors in design

One of the primary requisites in the design of this type of equipment is adequate power at the spindle. Existing machine tools of a size suitable for economic boring are underpowered for this operation, where a machine comparable in size to a 10 in swing centre-lathe requires up to 40 h.p. at the spindle. Any appreciable reduction in horsepower does not allow full advantage to be taken of the tungsten-carbide tipped tools. A secondary consideration arising from the use of high power is ample stiffness throughout and complete absence of vibration at all speeds.

To obtain the optimum rate of penetration the spindle speed and boring-bar



1, Headstock: 2, Chuck: 3, Workpiece: 4, Bored hole: 5, Roller-steady: 6, Oil-seal: 7, Oil pressure head: 8, Oil inlet: 9, Boring-bar: 10, Boring-bar headstock: 11, Swarf exit: 12, Carriage

Fig. 1. Semi-diagrammatic layout of a deep-hole boring machine

feed must be matched in relation to each other; the precise cutting condition necessary can only be realized by this means. This matching can best be achieved by the use of infinitely variable speeds and feeds that can be altered during machining. A variation in feed as small as 0.0002 in/revolution can cause an unsatisfactory chip to be produced and build up to form a blockage in the boring bar. This is a particularly important point, as current German practice is based upon using the correct speed and feed combination to produce the desired chip form, in preference to using a chip-breaker groove on the tool, when machining free-cutting steels.

The actual production of a satisfactory bore after the various mechanical considerations have been met is entirely dependent upon the cutting oil. In use, this oil has to perform three distinct functions which are:

1. Assist swarf removal.
2. Help the natural chip-breaking characteristics of the material.
3. Maintain the workpiece at a constant temperature throughout the boring operation.

To perform these three functions, the oil must be blended to suit the characteristic of the material being machined and not merely selected from a standard range of oils. In the light of current practice, the minimum pressure for satisfactory operation (depending, of course, upon the ratio in terms of diameter between oil entry and exit orifices) is between 600-700 lb/in² for the medium range of diameters rising to 1,000 lb/in² for small holes.

Boring-tool design

As mentioned in the introductory remarks, there are three principal methods of producing a deep hole by this technique. These are shown schematically in Fig. 3 and the upper diagram refers to what is known as the Beisner head, developed by Dr. Karl Beisner of Ruhrstahl A.C. Heinrich-

shutte for gun-boring operations to replace the D-bit type of cutter. In common with all the boring heads, the Beisner pattern makes use of two tungsten-carbide tipped rubbing pads along the body of the tool. One of the pads is 180 deg opposed to the plane of cutting, and the other 85 deg to the plane of cutting, beneath the cutting edge. The cutting oil is forced around the outside of the boring bar, passing through two or more flutes in the head, returning with the chips through an irregular-shaped orifice in the face of the tool.

With reference to Fig. 7 it will be seen that the point of the solid-boring tool is offset from the axis of the cutter head so that there is at least a reasonable cutting speed at the small diameter. This type of head—integral with the boring bar—has been used for producing holes as small as $\frac{1}{4}$ in diameter to a depth of 6 ft. For large pierce-boring operations it is satisfactory up to 4 in diameter. The trepan-boring method, whereby a solid core of material remains after machining, is illustrated in the centre diagram, Fig. 3. This process is not to be recommended, however, for bores less than 1 $\frac{1}{2}$ in diameter if the machine has sufficient power to allow a high rate of penetration by the Beisner system. The reason for this will be appreciated when it is seen that a very small chip has to be produced to pass between the core of the stock and the inside diameter of the boring bar.

The third method, shown at the bottom of Fig. 3, employs a counter-boring head; this type of tool can bore intermediate diameters after using a standard-size Beisner head. These counterbores are normally made from

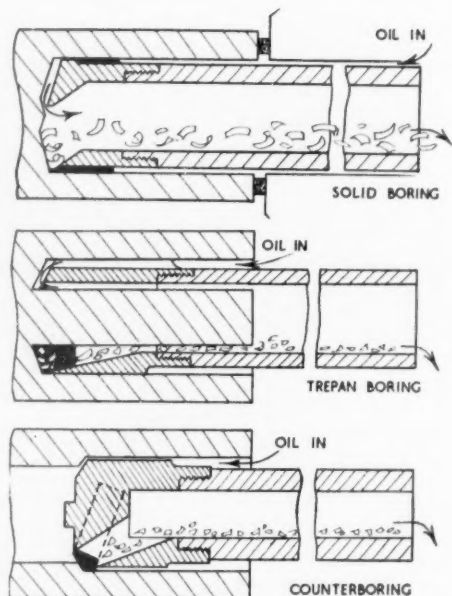


Fig. 3. The three principal types of cutter-heads, basically of German origin and developed by Heller

$\frac{1}{2}$ in diameter, where the cutter has a range of adjustment from a minimum of 0.004 in to about 0.1 in on diameter. In the larger sizes, a counterbore of 4 in nominal diameter has a cutting range from a minimum of 0.01 in to 0.7 in on diameter. It will be seen that the counterbore can be used in the same manner as a Beisner head for machining blind holes, as distinct from trepan boring where the bored hole must extend the full depth of the component. In the larger diameters the rubbing pads are detachable to effect a considerable economy as their life is greater than the cutting tip. The main advantage of these methods, apart from the increased rate of penetration as compared with the D-bit, is that the chips are conducted through the boring bar and do not abrade the surface of the bore with consequent loss in surface finish.

An example of trepan boring a relatively small diameter is shown in Fig. 4. This operation is carried out on a slightly modified No. 4 Gisholt turret lathe with portable boring equipment. The original method of machining the component consisted of drilling and reaming operations. The trepan-boring technique requires an average machining time of only 3 $\frac{1}{2}$ min, with the added advantage of a greatly superior surface finish. The component is supplied as a nickel-chrome steel forging (Spec. S11) measuring 13 $\frac{1}{2}$ in long and is through-bored 1.2 in diameter.

Preliminary machining consists of centering and rough turning the O.D. of the circular portion of the forging, and terminates at one end in a 20 deg included angle taper for a metal-to-metal oil seal in the pressure head, Fig. 5. The workpiece is loaded in a counterbalancing fixture, to eliminate vibration, and is held by a latch-type

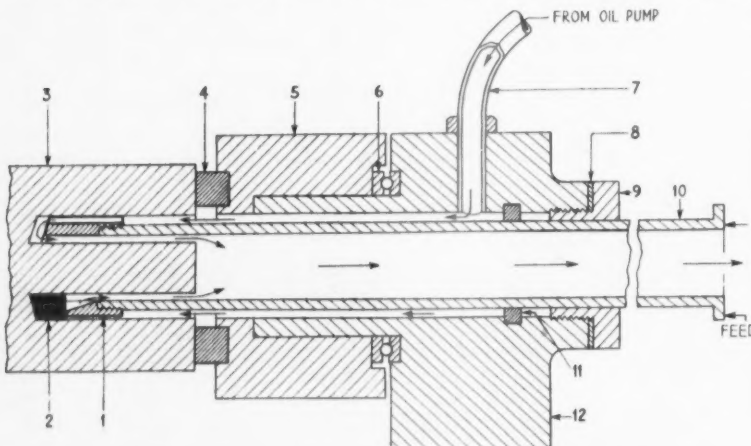
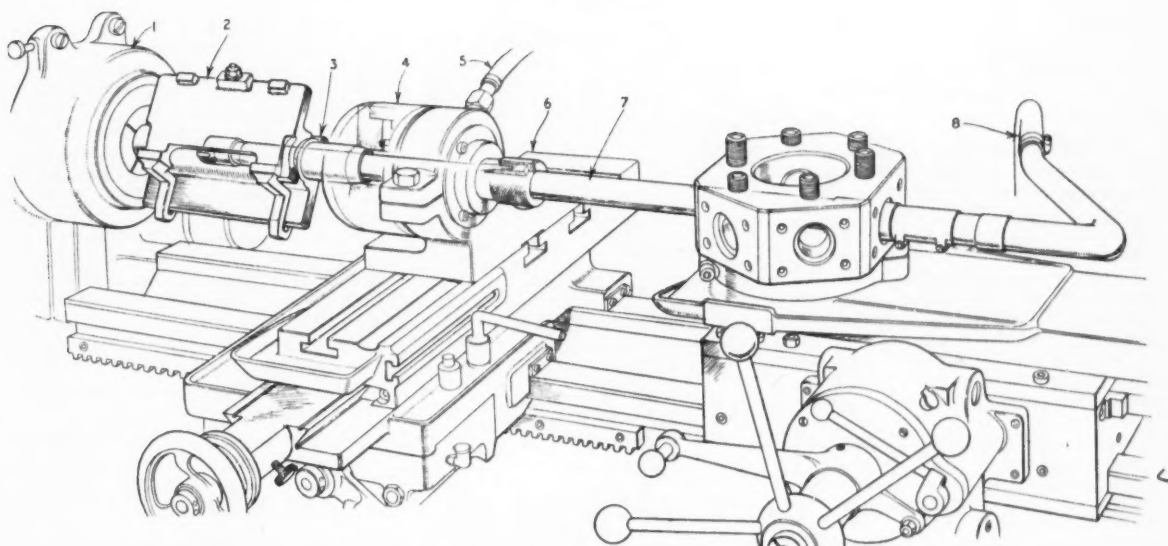


Fig. 2. The principle of a typical pressure head. Oil is fed in at between 200/1000 lb/in² pressure in the stationary part of the unit and directed through the rotating member and around the outside of the boring-bar



1, Headstock; 2, Counterbalancing fixture; 3, Workpiece; 4, Pressure-head; 5, Oil inlet; 6, Adjusting nut; 7, Boring bar; 8, Oil exit pipe
Fig. 4. Trepan boring on a modified No. 4 Gisholt turret lathe

clamp. The assembly is inserted into the collet-chuck of the machine and the pressure head, mounted on the cross-slide, is brought into contact with the free end of the part and is locked in that position. As seen in Fig. 5, the forging is held in the headstock of the lathe and located inside a length of thin-wall steel tubing, split at the collet end to allow the workpiece to be gripped, and sealed at the opposite end so that the cutting oil must return through the boring bar.

Sultran B1 (Vacuum Oil Co. Ltd.) cutting oil is used, and this is delivered to the cutter head at a pressure of 200-250 lb/in² at the rate of 20 gal/min. About 250 gal of coolant is in use, and the oil returning from a circuit is filtered through a wire-mesh basket before re-circulation. Despite the seemingly short bore, the rise in oil temperature becomes very noticeable after a few components have been bored. This is indicative of the cutting pressures entailed in comparatively high-speed trepan boring, and the higher-than-normal load on the headstock bearings of a standard machine.

Although the trepanning heads, Fig.

6, are based upon the original German design (Gebrüder Heller, Bremen-Mahndorf*) much of the credit for satisfactory operation is due to the Martin-Baker Aircraft Co. Ltd., who are using the equipment and have redesigned the tip contour. To provide sufficient power at the spindle, a 10 h.p. motor has been fitted to the machine. Depending upon the hardness of a batch of forgings, a spindle speed range from 800-1,000 r.p.m. is used in conjunction with a feed rate of 0.005 in/rev. Providing that the cutter is not damaged in use, a total service life of a head is between 200-300 ft of boring.

Much of the recent development work in deep-hole boring has been concerned with the production of the smaller sizes of hole ranging from 1/4 in to 1 1/2 in diameter, with the result that a number of commercially acceptable machines are in production and other designs are ready for manufacture. In this sphere, Heller have achieved a series of remarkable successes with their cutter-heads with a degree of consistency that eliminates the possi-

bility of freak results. The following figures can be regarded as typical of this development work on a production basis.

Material 54-60 ton tensile stainless steel, normalized

Solid boring 1.38 in diameter, tolerance ± 0.0004 in, with Beisner-type head, in lengths of 37 1/2 in

Feed rate 9.84 in/min

Material 85-92 ton tensile stainless steel

Solid boring 1.38 in diameter, with Beisner-type head, in lengths of 37 1/2 in

Feed rate 5 1/2 in/min

Material 38-44 ton tensile steel

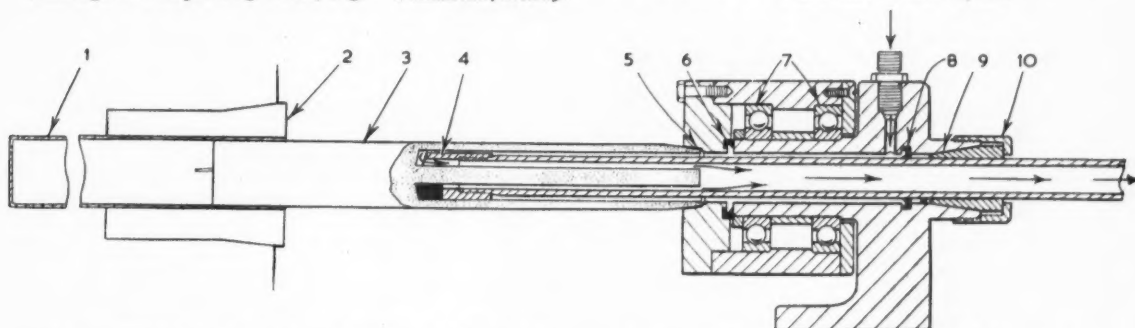
Solid boring 0.47 in diameter, with Beisner-type head, in lengths of 23 1/2 in

Feed rate 9.45 in/min

Material 45-55 ton tensile stainless steel

Solid boring 0.326 in diameter, with Beisner-type head, in lengths of 27 1/2 in

Feed rate 5.98 in/min



1, Steel-tubing; 2, Collet; 3, Workpiece; 4, Cutter head; 5, Oil seal; 6, Rubber seal; 7, Ball-bearing; 8, Rubber seal; 9, Hardwood bush; 10, Adjusting nut
Fig. 5. Section through the Heller-designed oil pressure-head for the trepan-boring operation. A steel tube is fitted inside the collet to prevent oil from discharging through the headstock spindle

*Heller boring heads are handled in this country by Wickman Ltd., Coventry.

Material 70 ton tensile stainless steel
Trepan boring 3-15 in diameter, in lengths of 18 ft 6 in
Feed rate 5½ in/min

Although the depth bored in some of the foregoing examples is not very great due to component design, a similar rate of penetration could have been obtained for solid boring up to a depth of at least 15 ft. With work of this nature, the design of the oil pressure head can become a critical factor, and although a number of designs, a selection of which are shown in Figs. 8 to 11, are available, the principle of operation is similar to that shown in the schematic diagram, Fig. 2.

Heller have developed a number of standard heads of the pattern illustrated in Fig. 8 which are suitable for insertion into an existing housing or

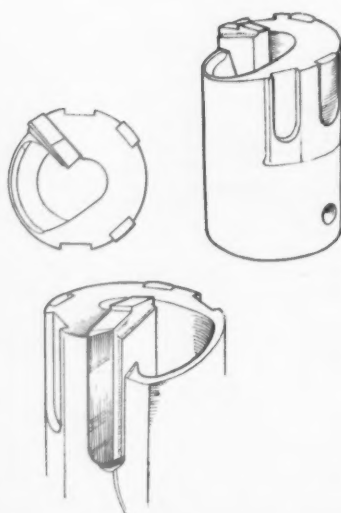


Fig. 6. Details of the trepanning cutters. The heads are manufactured from a high-tensile steel body with a single tungsten-carbide tool bit

Pressure head design

The V.D.F. group of companies (Vereinigte-Drehbank-Fabriken) makes a number of boring machines in two distinct ranges, the V.D.F. range manufactured by Heidenreich and Harbeck, of Hamburg 33, and the Boehringer range by Gebrüder Boehringer GmbH, of Goppingen*. Fig. 12 shows the rear view of model B3 V.D.F. machine. In this type of machine the pressure head, which has an integral pressure gauge, is traversed almost up to the work-piece, and the carriage clamped to the bed of the machine. With the main assembly rigidly held to the bed, the pressure-head component is advanced to make contact with the end of the workpiece by rotating a feed-nut by means of the spoked wheel.

One of the difficulties associated with deep-hole boring is the maintenance of the cutting oil at a constant low temperature to prevent linear expan-

*Both the V.D.F. and Boehringer boring machines are handled in this country by Sykes Machine Tool Co. Ltd., Terminal House, Grosvenor Gardens, London S.W.1.

sion of the workpiece. This aspect of machining is important as, for all practical purposes, the workpiece is held between two rigid members—headstock and pressure head—and any appreciable rise in temperature must result in an increase of pre-loading on the headstock bearings of the machine. In certain instances, the thrust mechanisms have been overloaded to the extent of causing a bearing failure.

The extent of the increase in loading due to expansion can be ascertained from the following hypothetical example:

Temperature of workpiece at commencement of boring, 68 deg F
 Maximum temperature of workpiece during boring, 140 deg F
 Temperature rise, 140-68=72 deg F
 Nominal coefficient of linear expansion for steel, 0.0000636 per unit length per deg F



Fig. 7. A Heller solid-boring head developed from the Beisner system

similar type of supporting body. This type of head in its standard form has a plain rubber sealing ring between the workpiece and the rotating member the face of which is formed with a square-section groove for retaining the sealing ring.



Fig. 8. One of the largest types of Heller pressure-head. There is a range of this type of unit

Nominal coefficient of linear expansion for aluminium 0.0001234 per unit length per deg F

On this basis, a steel component 9 ft long would increase in length by 0.0494 in and an aluminium component

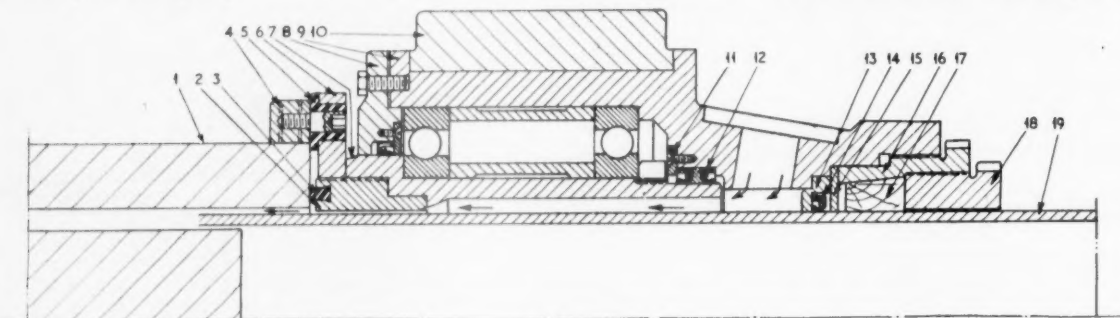
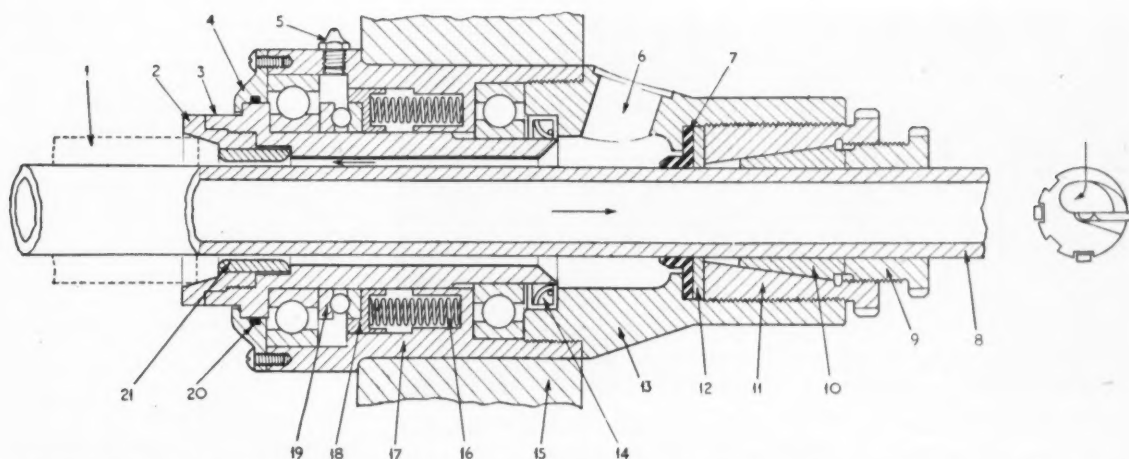


Fig. 9. Section through a Wagner pressure head. The oil-seal adapter is fitted to the outside diameter of the workpiece and held by through-bolts to the pressure-head component



1, Workpiece: 2, Metal seal: 3, Driving-bush assembly: 4, Cover plate: 5, Oil nipple: 6, Oil entry: 7, Rubber seal: 8, Boring-bar: 9, Lock screw: 10, Vibration-damping bush: 11, Seal lock screw: 12, Seal washer: 13, Body: 14, Oil-seal assembly: 15, Housing: 16, Compression spring: 17, Main body: 18, Thrust-absorption ring: 19, Thrust bearing: 20, Rubber seal: 21, Boring-head bush

Fig. 10. Details of a German spring-loaded pressure head for absorbing the expansion of a workpiece

by 0.0959 in. The two principal methods of overcoming the effect of this expansion are, first, the use of a larger flow of cutting oil when this is practicable, entailing the circulation of a much larger volume of oil, or the addition of refrigerating equipment in the oil reservoir to ensure that the oil is cooled before re-use and, secondly, a means of absorbing the expansion within the pressure head.

When the second method is the only practical solution, it is advisable to use a pressure head of the pattern illustrated in Fig. 10. Originating in Germany—the actual source unknown—it is suitable for either rubber-to-metal seal or entirely metal-to-metal contact between the head and workpiece. The body is made in two parts (13 and 17) and the assembly as a whole is fitted into the pressure-head housing. At the forward end, a metal-to-metal seal (2) is in contact with the workpiece. The seal is bushed internally (21) to suit the cutter head; the assembly is fitted into the main-bearing bushing (3) supported by ball bearings which are a press-fit around the outside diameter of the bush. A thrust bearing (19) is introduced between the ball-bearings and is fitted into a thrust-absorption ring (18), which has a number of pockets for retaining helical compression springs.

In use, the pressure-head is brought into contact with the workpiece in the normal manner to provide a seal, and in so doing, the main bearing-bush assembly is slightly compressed against the rear surface of the spring seatings. Any extension of the workpiece caused by heating is absorbed with the minimum increase of headstock bearing pressure.

Machine design

With the exception of the Carlstedt equipment mentioned later in this article, existing boring machines have the inherent fault of possessing stepped speeds and feeds. Because of this, the results obtainable are to an extent limited by the inevitable compromise existing between fixed ratios. Even with this handicap however the rates of penetration by this technique are at least double those obtaining at the present time by conventional methods, with the additional advantage of a markedly superior machined surface.

A good example of a machine conversion is illustrated in Fig. 13. This machine, originally a Loewe horizontal boring machine, has been converted for deep-hole boring by Heller, and is installed in their Bremen-Machindorf works. In this instance the cutter head rotates instead of the workpiece and the pressure head is integral with an

extended-length carriage which serves as a table for the workpiece. The workpiece is held by transverse strap clamps and an end clamp attached to the pressure-head housing.

As illustrated, the machine is being used for boring an 18-8 high-tensile steel billet. A Beisner-type solid boring head 1.58 in diameter is used for boring a hole 20 in deep at a feed rate of 7.7 in/min. While the set-up is a temporary method for boring eccentrically located holes, a turret-type fixture could be used for long run production.

The V.D.F. machines, to which passing reference has already been made, are basically smaller versions of the Boehringer range. As will be seen from the various illustrations, the headstock is similar in design to the standard V.D.F. lathe but has a greater range of speeds and feeds. Another feature—but also common to the Boehringer range—is the duplication of primary controls on the pressure-head carriage and boring-head carriage. This form of duplication is an essential factor in the design of this class of machine where, owing to the use of cutting oil at high pressure, coupled with a high rate of tool penetration, it is necessary for an operator to have full control of the machine while he is on either side of the pressure head.

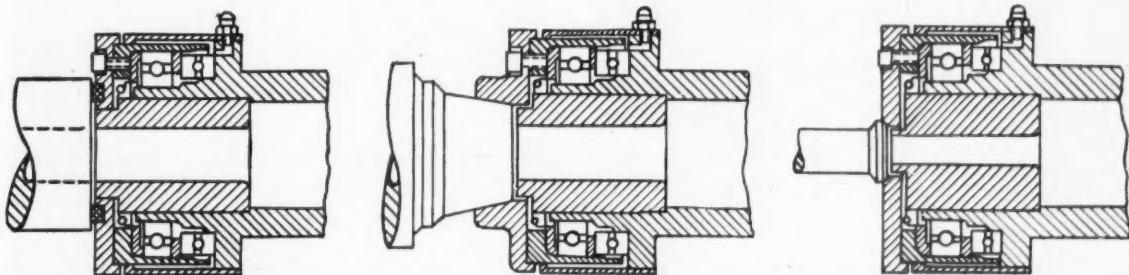


Fig. 11. Three methods of workpiece-to-pressure head seal used on V.D.F. machines. (Left) Conventional type of rubber-to-metal seal. (Centre) The preferred type of slow-taper metal-to-metal seal. (Right) Acute angle profile seal to suit component design. It is doubtful whether this type of seal is entirely satisfactory

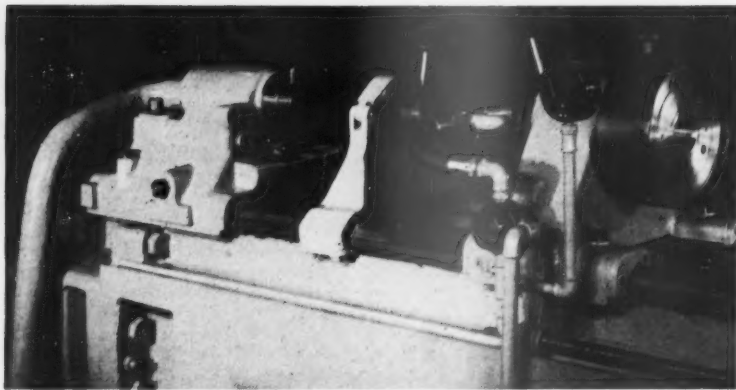


Fig. 12. Rear view of model B3 V.D.F. deep-hole boring machine, showing the adjustable position pressure head

One of the methods used for securing a boring-bar to the boring head is shown in Fig. 14. Here, the end of the bar has a circular groove which is located by a sliding lock clamp. This type of fixing, of course, is only suitable for a stationary boring bar. When a rotating bar is used, it is customary to have either a collet-type lock or a flange fitting with through-bolt clamping. The oil return system shown in the same illustration has not proved entirely satisfactory because of constriction and many changes in direction; for this reason it has now been modified to allow a more direct flow.

The Boehringer machines, Fig. 15, are in the medium-to-large capacity class and can be supplied in two basic versions. The first design, Model B5 (as illustrated), has a stationary boring-bar, while the second type (Model B5B) has a combined rotary boring-bar. In addition to the basic design, two methods of headstock drive are available; the first uses a standard 20 h.p. motor mounted on a baseplate with tensioning brackets for a vee-belt drive, giving a range of 18 spindle speeds in a progressive ratio of 1:25. When this system is used, a double pole changing three-phase motor, or a two-

speed D.C. motor can be supplied. The second method of primary drive consists of a flange-drive motor giving a speed range of from 11.8 to 600 r.p.m. in the same progressive ratio.

Each method of driving has an adjustable multi-plate clutch which can be operated from the feed box, the boring head or the saddle. When the drive is disengaged, both the main-drive motor and coolant pump are stopped simultaneously. The power for the feed motion is obtained from the headstock spindle through a reversing-gear mechanism. A reversing-gear unit has been incorporated in the feed mechanism in order to carry out boring or reaming by the draw-cut principle: a range of 20 feeds can be obtained in a progressive ratio of 1:40 from 0.00047 in to 0.335 in/rev. Quick adjustment and rapid return of the boring head can be made at speeds between 3½-6½ ft/min.

In its standard form, the pump unit gives a coolant pressure of up to 284 lb/in², although equipment adapted to give pressures up to 711 lb/in² can also be supplied. This machine has a boring capacity of 4½ in diameter when using a Beisner or D-bit type head, and 7½ in diameter for trepan boring. Available in a number of bed-lengths,

the smallest machine can handle work up to 3 ft 3 in in length and the largest, work up to 19 ft 8 in in length.

The Carlstedt* range of deep-hole boring machines mentioned in the November issue of *Automobile Engineer* (pages 481-2) represents the latest practice in the design of this class of equipment. A typical result claimed by the manufacturer is:

Material High-alloy steel

Solid boring Beisner - head method, ½ in diameter, in 6 ft lengths

Feed rate 6½ in/min. Maximum eccentricity at the break-through end, 0.0012 in. Tool life between regrinds, 65-78 ft of penetration.

A feature of this machine (Figs. 16 and 17) is the ready accessibility of the headstock for loading the work at the rear of the machine. This arrangement, coupled with a pneumatic work-holding chucking system, allows the incorporation of a roller-type conveyor loader where the component to be bored is allowed to roll into the loading position on the machine immediately the machined component has been released from the fixture into the delivery chute.

Vertical boring

Equipment suitable for parts of shorter length or forgings of complex shape is built by Hille-Werkzeugmaschinen GmbH, of Witten-Annen, Ruhr, Germany†, on the unit construction principle for battery operation. A typical double-spindle machine of this type is shown in Fig. 18. This class of machine is used for boring either batch-production or quantity-production parts in the inverted position. The part to be drilled or bored is held in a chuck or fixture above the cutting tool. As illustrated, the machine is equipped with a high-pressure oil head and a Beisner-type boring tool. Here, the oil is pumped upwards through the pres-

*Now handled in this country by Wickman Ltd., Coventry.

†Handled in this country by the Rockwell Machine Tool Co. Ltd., Welsh Harp, Edgware Road, London N.W.2.



Fig. 13. Another Heller pressure head is integral with the carriage. Here the boring-bar is rotated instead of the workpiece

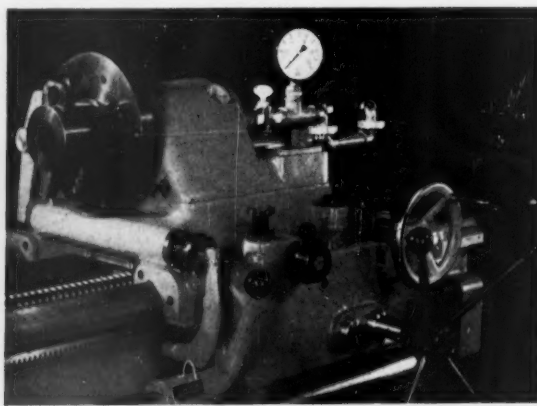


Fig. 14. Method adopted by V.D.F. for securing a stationary boring-bar

sure-head and around the outside of the boring-bar, returning through the centre of the bar and escaping through the cut-out shown in the fixed-head fitting in the swarf tray. Known as type BST 25.2, a performance figure claimed for a machine of this type is as follows:

Material Steel, composition C 0.27, Si 0.28, Mn 0.3, Cr 1.5, Ni 4.5, Mo 0.5, T 4.5, V 0.2

Boring diam. 0.669 in with Beisner head

Boring depth 5.102 in

Output Floor-to-floor time 2.9 min per part per spindle.

The advantages of this type of machine are the relative ease of production once the machine has been set for an automatic cycle, and the ready incorporation of special work-handling jigs or fixtures to ensure repetitive alignment. A disadvantage, from the point of view of deep-hole drilling, is that it is not equipped with infinitely variable speeds and feeds.

Cutting oil

In deep-hole boring, success is perhaps more dependent upon the active co-operation of the cutting fluid than in any other metal-removing process. Because of the nature of the operation the oil is, in effect, the third of the basic requirements, of which the other two are machine and cutter head. There are a number of requirements that all cutting oils must meet. These are, briefly, heat removal; ability to wet the metal efficiently so that cooling may be as effective as possible; maintenance of tool life, which is dependent upon viscosity in order that the oil will remain at the cutting edge while still allowing easy circulation and swarf removal; anti-welding properties; fluid stability with freedom from separation in use; and a minimum tendency to fuming.

This class of boring creates an arduous set of conditions for the fluid and, because of this, neat oil of the active type should be used. The terms "active" and "inactive" oils, refer to

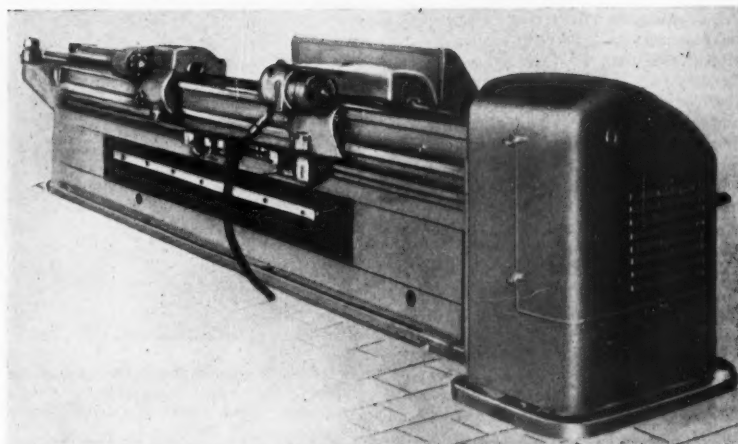


Fig. 16. The basic version of the Carlstedt (Swedish) deep-hole boring machine. The smallest version can be used for boring holes $\frac{1}{4}$ in diameter

the chemical activation of certain constituents in the oil which react on the surface of the work to improve cutting conditions. These resultant reactions are affected by the pressure and temperature of the work at the point of cutting and with the susceptibility of the workpiece to the chemicals.

The purpose of these compounds is to prevent the adhesion of swarf at the point of the tool, and subsequent generation of a roughened surface on the workpiece. Although fatty oils, which are of the polar type, have a good effect upon boundary friction, the application of mineral fatty oils has diminished over the years in favour of the "extreme-pressure" (E.P.) oils. These oils possess greater anti-welding properties and contain further additives to allow them to withstand high pressures between the workpiece chip and the tool. Following the discovery of the application of free sulphur to cutting oils came the steady and continuing development of E.P. oils. Fluids of this type have a still greater tendency to react at the point of cutting, giving rise to the formation of a strong chemical film between the two surfaces. The additives in present day

E.P. oils to create this film contain either or both sulphurized and chlorinated properties. Apart from laboratory tests combined with actual performance tests, the only short cut in testing procedure is to experiment with a selection of fluids containing known additives and to vary the percentage of these additives until the desired degree of stability in production is achieved.

Chip formation

One of the primary functions to be established in the selection of cutting oils is the mechanism entailed when a metal is machined. By the use of contemporary photographic and metallographic equipment three basic chip forms have been established and these are the result of two fundamental processes. These comprise a shearing process whereby the initial chip form is created, followed by the movement of the chip up the face of the tool. It is now generally accepted that the three basic chip forms (equally applicable to boring) produced during machining are: continuous chip, continuous chip with built-up edge, and the discontinuous chip.

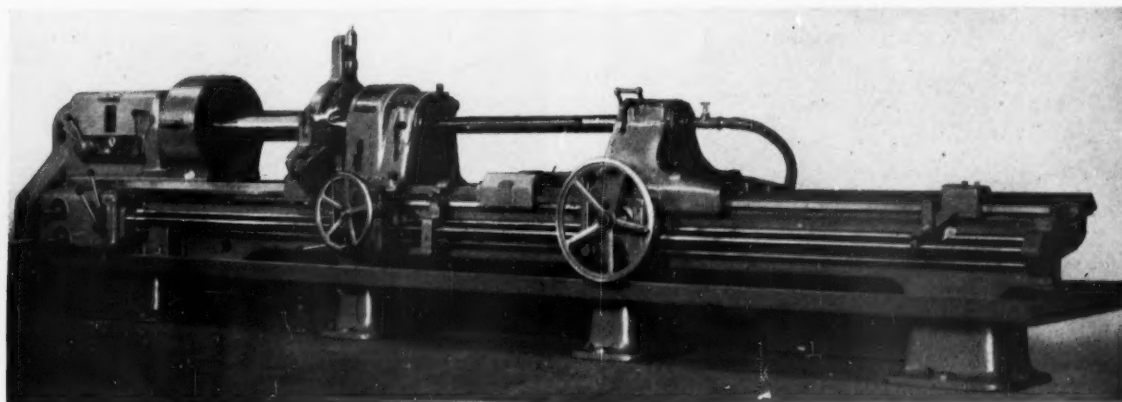


Fig. 15. The Boehringer model B5 boring machine with a D-bit head in position. This machine is also available in the combined rotating-bar version, when it is known as model B5B

A continuous chip is produced when the material is continuously deformed without rupture to flow smoothly up the tool face. The conditions responsible for this effect on ductile materials are a fine feed, high cutting speed, efficient cutting fluid and a keen cutting edge. The continuous type of chip with a built-up edge is most likely to take place with tough but ductile materials, and is brought about by high frictional resistance as the chip passes across the face of the tool, resulting in some of the material being sheared away before the body of the chip leaves the workpiece. Here, unless an active sulphurized oil is used, localized welding can take place. The conditions most likely to give rise to this effect are: ductile material, low cutting speeds, coarse feed and inefficient cutting fluid. When the built-up edge effect is of only slight proportions, it is not necessarily a fault, as it can serve to protect the tool. When possible, however, the heat of the molecular disturbance and the pressure should be used to force the surface-active bodies of the oil into the micro-fissures to form a solid metallic sulphide film of solid lubricant to prevent welding between chip and tool. With the discontinuous chip form, the shear stresses exceed the shear strength of the material and are the result of a small rake-edge on the tool, brittle material and low cutting speed.

To obtain a high grade surface finish, it is safe to give the order of preference of material chip forms as:—

1. Continuous chip with artificial breaking.

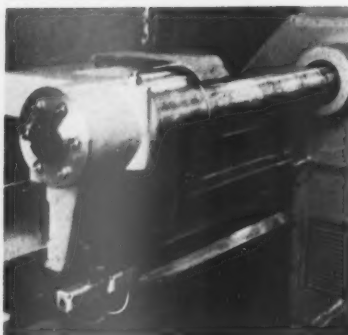


Fig. 17. Rear view of the Carlstedt machine showing the neat arrangement of the headstock which permits quick work-handling

2. Discontinuous chip with brittle material.
3. Discontinuous chip with low friction on ductile materials.
4. Continuous chip with high friction.

The principal factor known to have the greatest effect upon surface finish is the coefficient of friction between the tool and workpiece, and a reduction in this value will usually have the desired effect.

General conclusions

The technique of deep-hole boring, while apparently a relatively simple process, is in practice rather the reverse. Considerable power is required at the spindle for the mere act of removing the volume of metal which, in turn,

necessitates a structure capable of withstanding the thrusts generated. An equally important consideration is the provision of infinitely variable speeds and feeds or, at least, an infinitely variable feed. When a free-cutting material is being machined that does not require a chip breaker groove on the tool, extreme sensitivity of feed control to within 0.0002 in/rev can be important. In achieving this accuracy, the actual sensitivity of control on the part of an operator is also a factor; this requires a machinist of toolroom standard. To allow quick interchangeability, the design of a pressure head should be conceived as a unit for insertion into a housing integral with the machine. On the subject of cutting fluids, the position in this country is necessarily one of experiment. There is no apparent reason why a sulphochlorinated compound would not be eminently suitable providing that the question of sulphur-release temperature can be held within narrow limits.

Additional practical knowledge is also required to determine exactly the right chip form and hence the right tool angles. The only satisfactory approach to the compilation of accurate data of this kind is a methodical analysis. The process as a whole is a comparatively new method of production, and as such requires a certain amount of time to develop the most suitable means of incorporating it into an existing manufacturing programme. Whatever course is adopted, however, this technique will allow a considerable increase in the rate of production with the minimum of extra floor space.

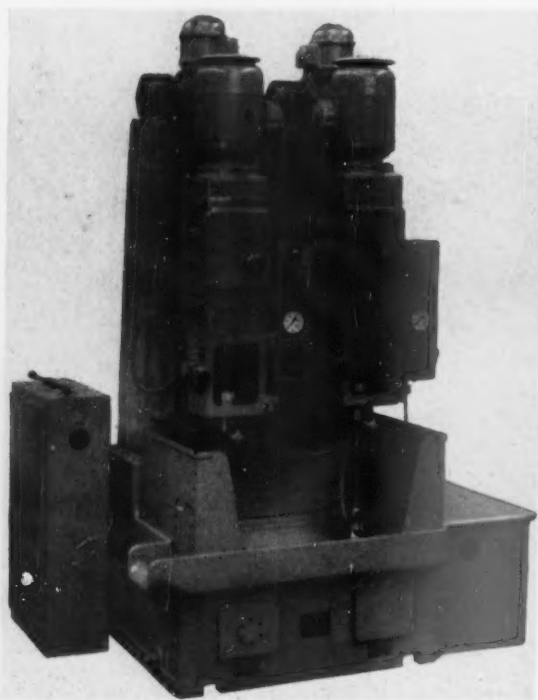


Fig. 18. A Hille deep-hole boring machine fitted with a Beisner-type gutter head for boring finished-machined spindles



Fig. 19. Details of the Beisner-head boring system fitted to the Hille machine



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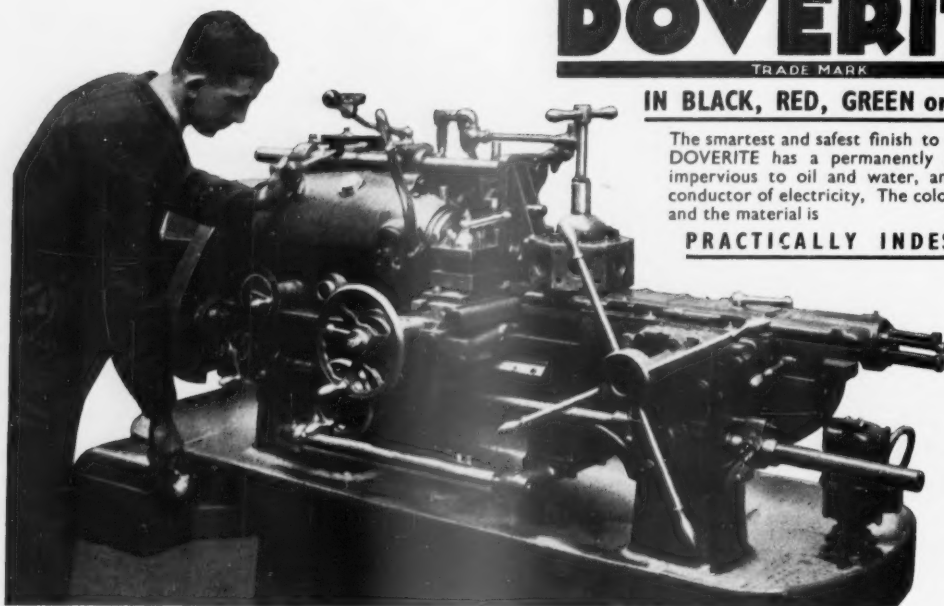
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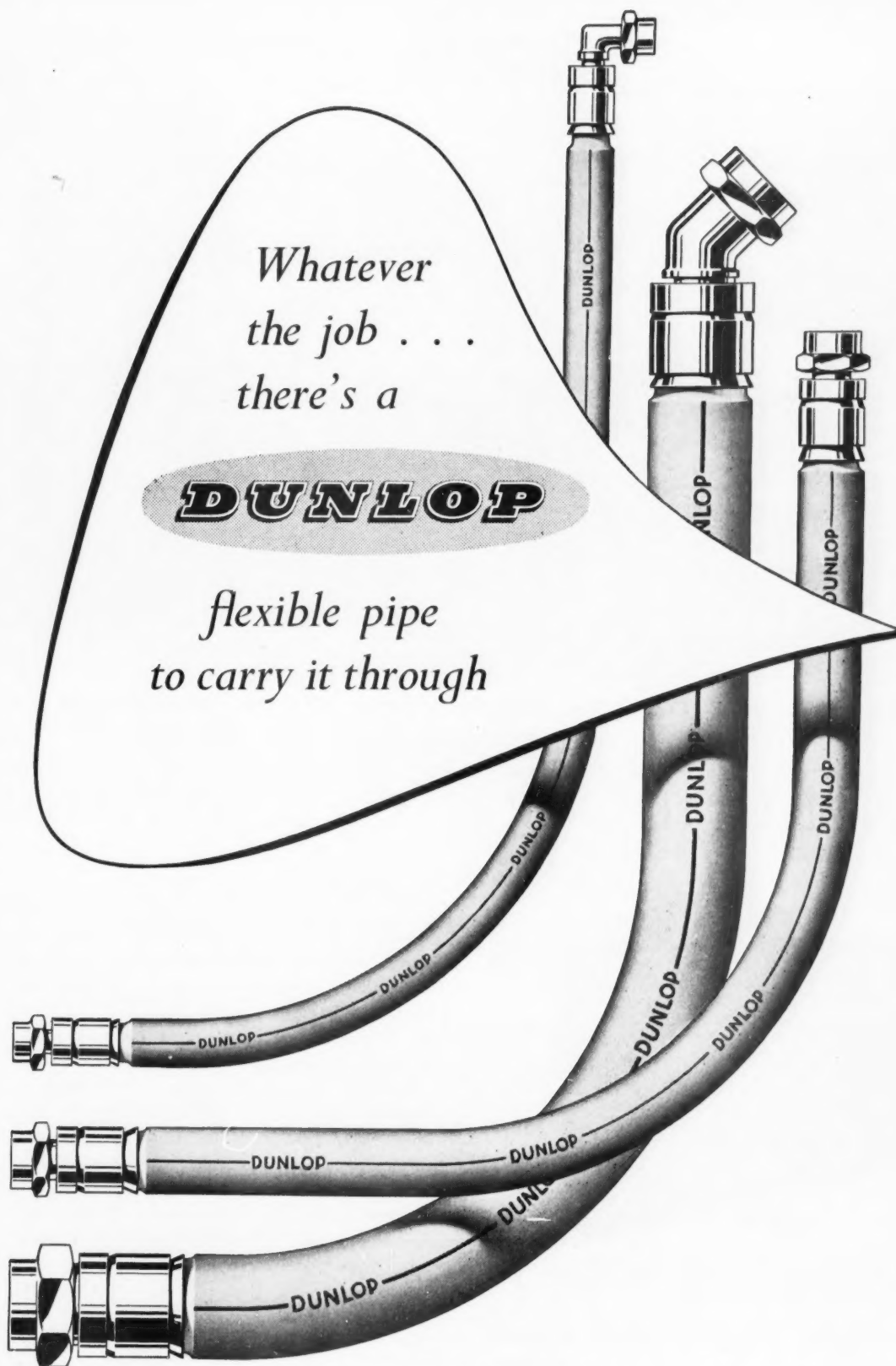


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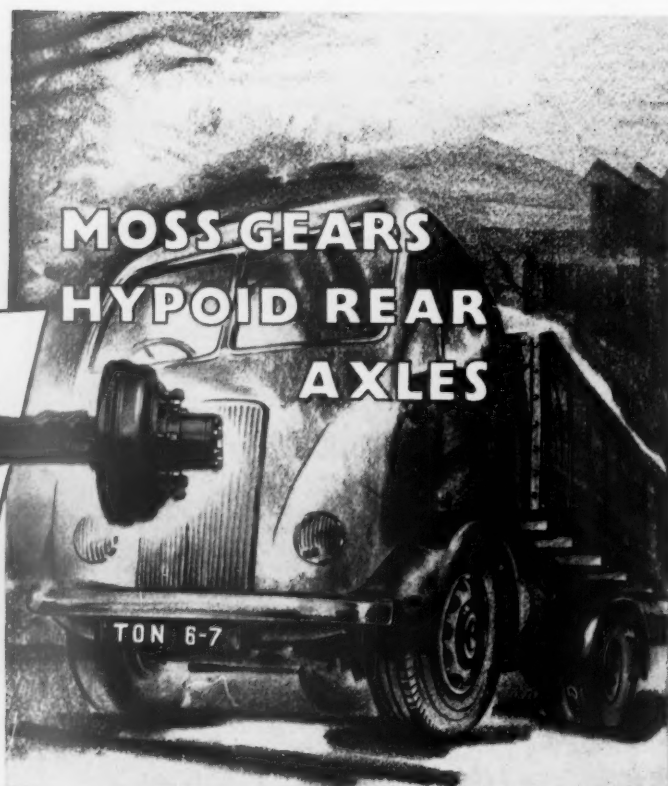


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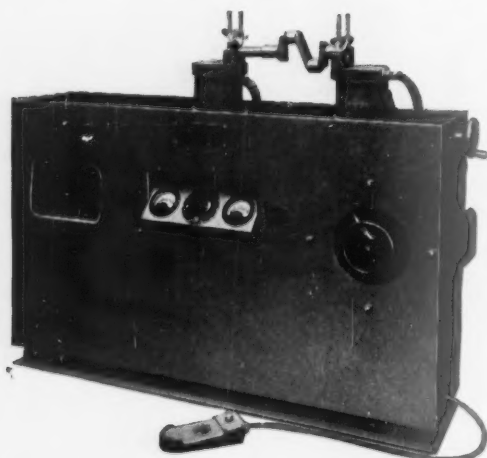
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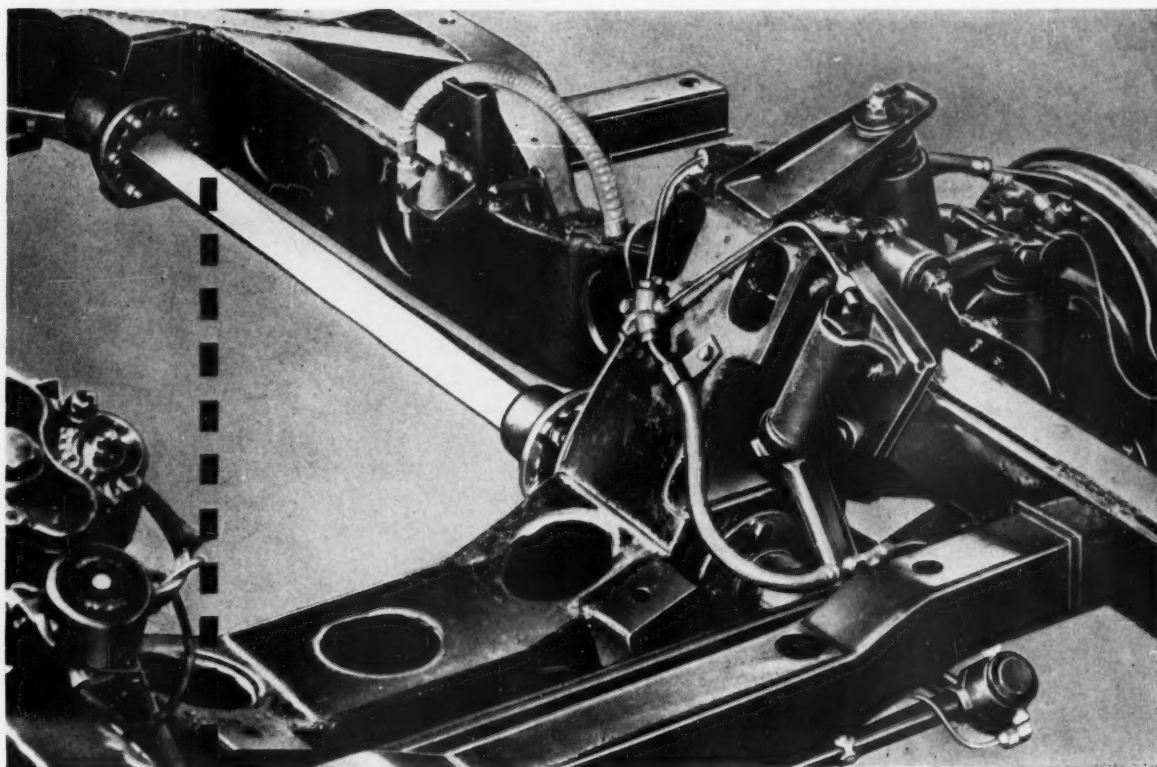
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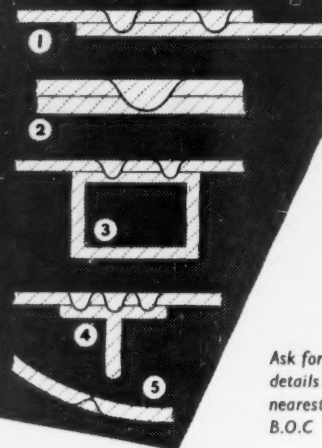
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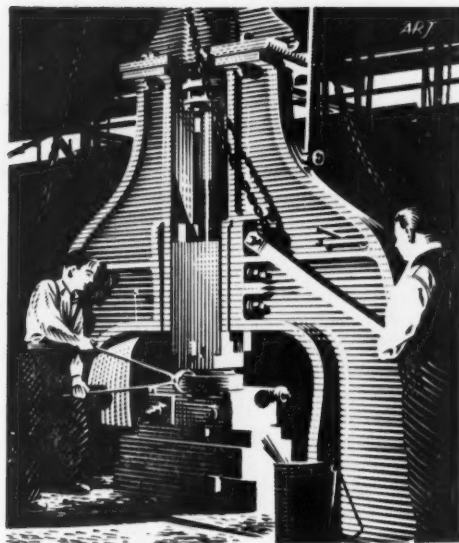
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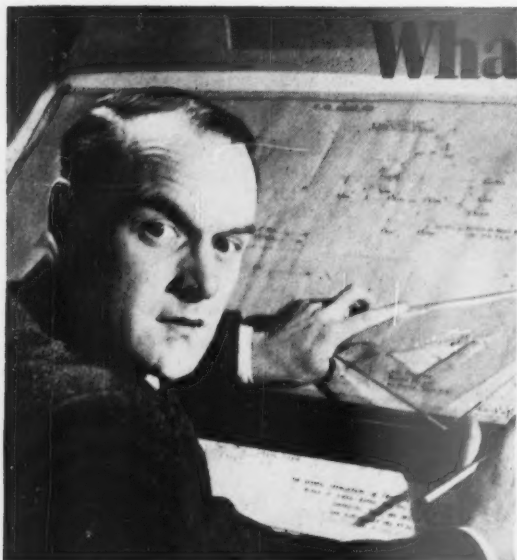
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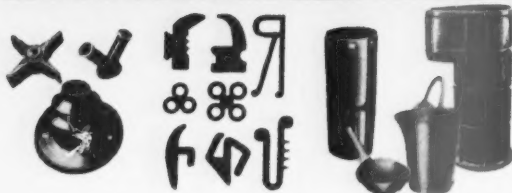
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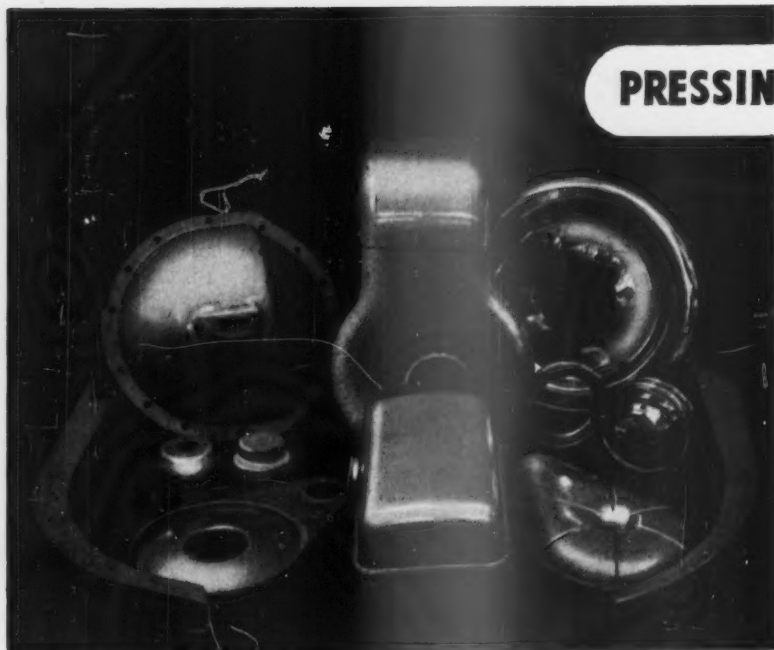
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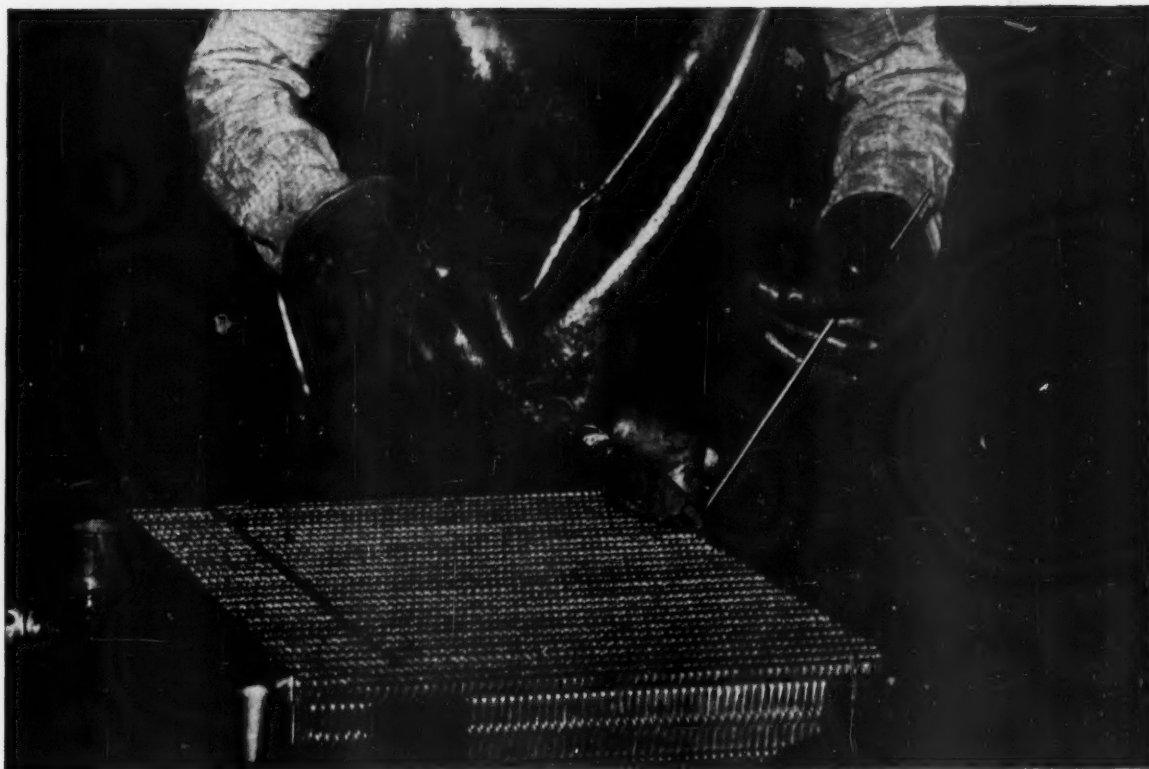
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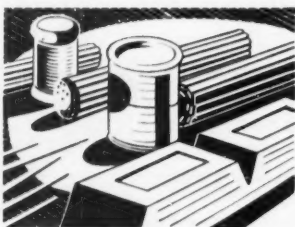
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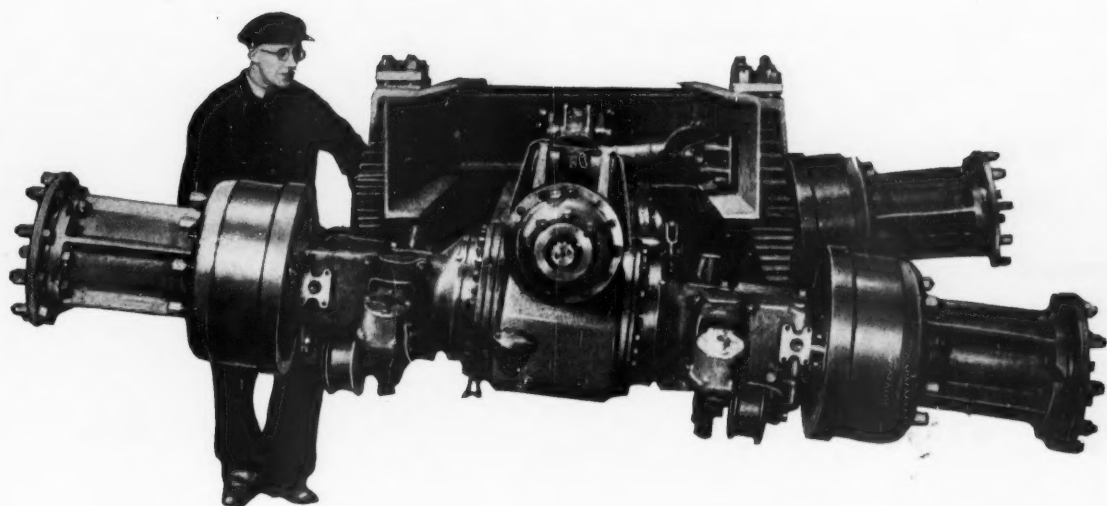
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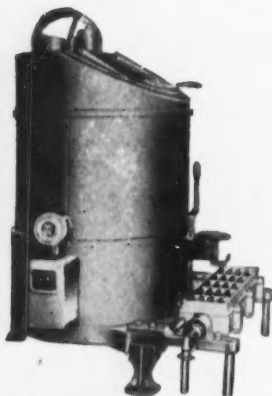
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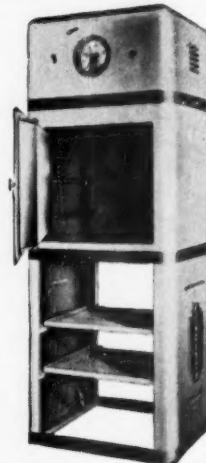
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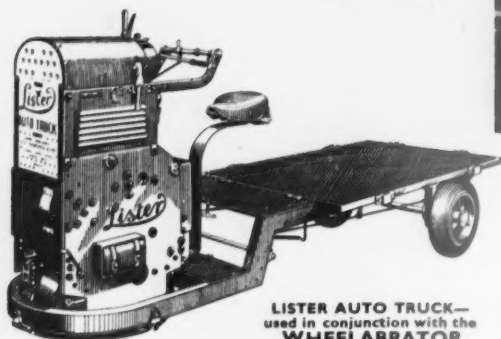
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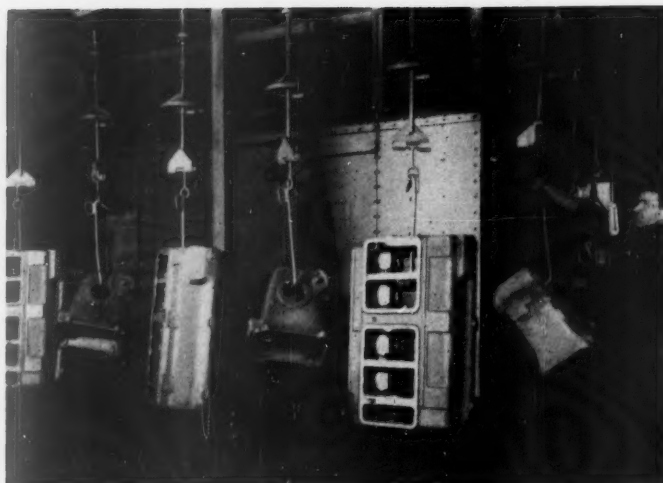
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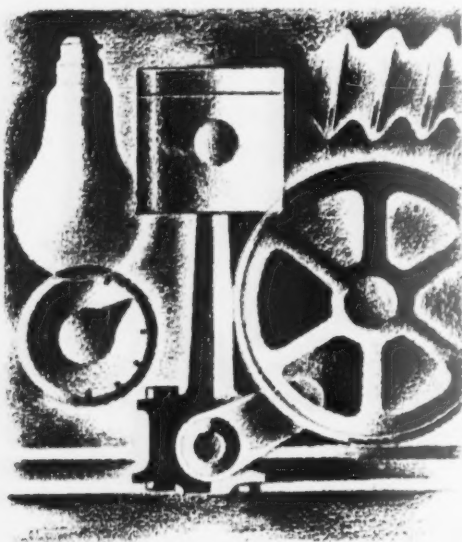
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W.22



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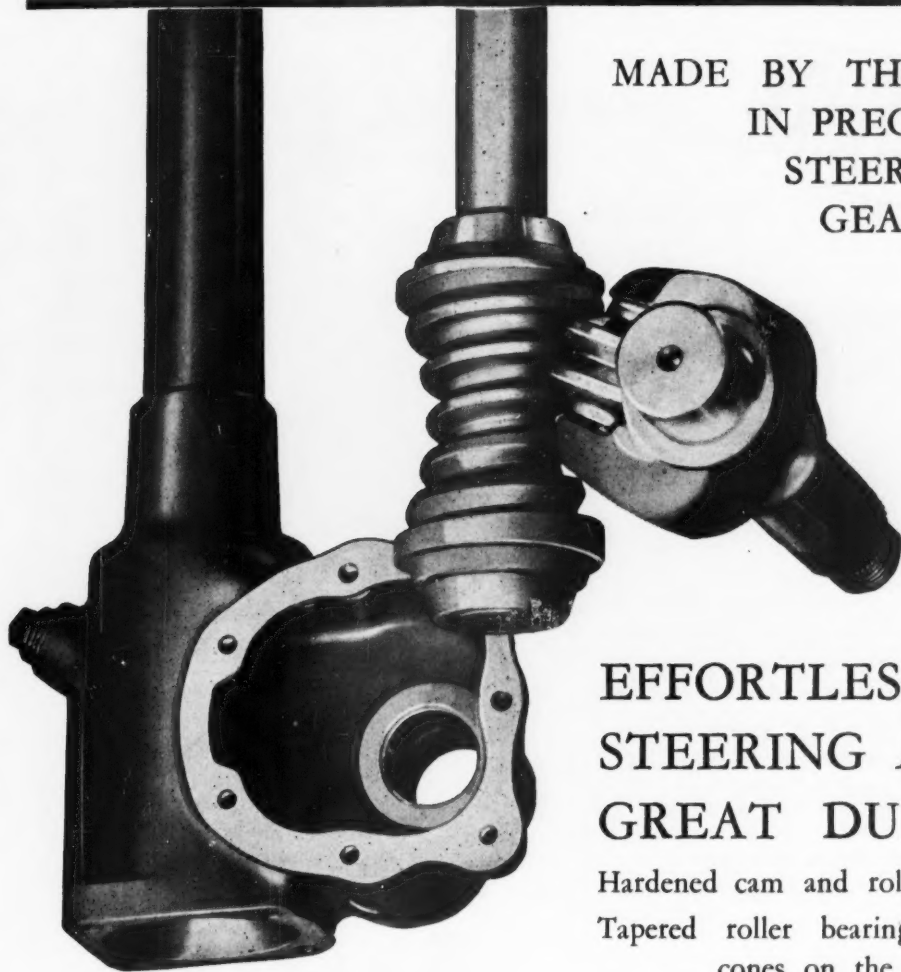
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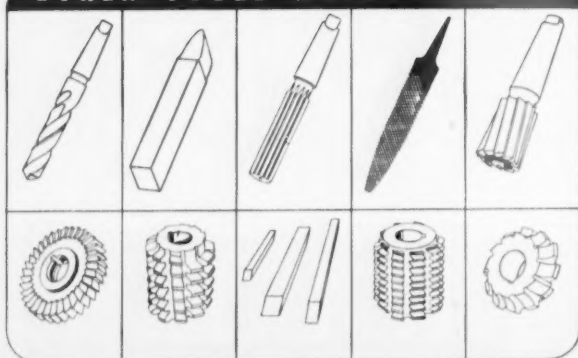
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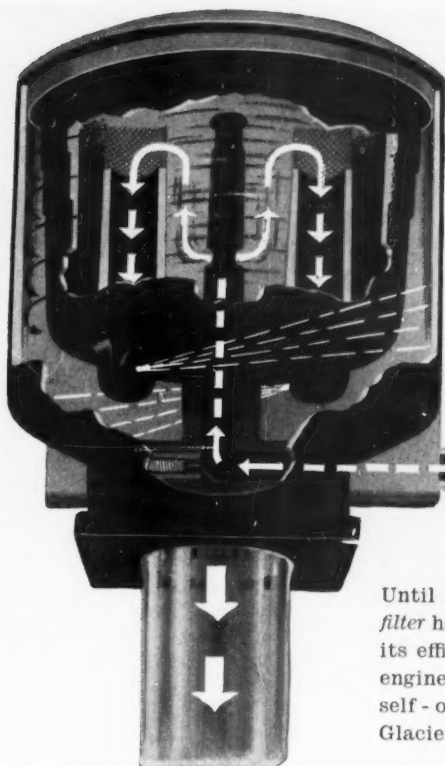
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Like many another technical advance, the idea is basically simple. In place of filter-packs and elements, the Glacier harnesses centrifugal force (at a thousand times the force of gravity) to do the job better. Look at the sectional diagram, and you will see how the oil itself drives a rotor which flings the dirt to the sides of the bowl, while the clean oil flows unchecked through the hole in the base.

THE TEST OF EFFICIENCY

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FOR VEHICLES &
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ENGINES**

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Ref. No.	Pipe Size	Union Type
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1124B	$\frac{1}{2}$ " O.D.	Solderless
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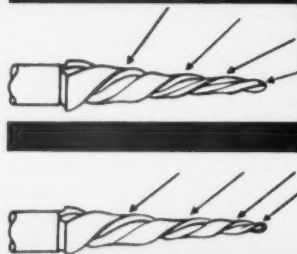
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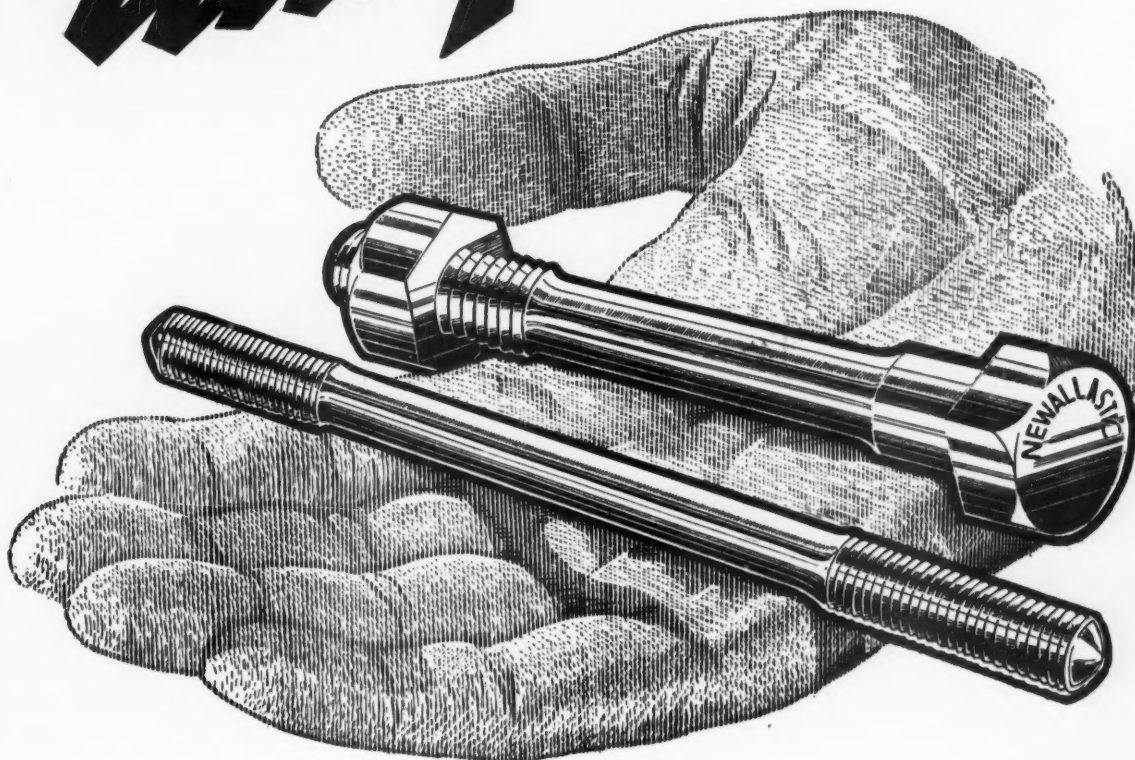
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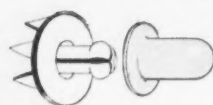
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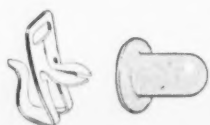
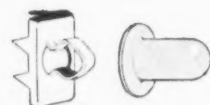
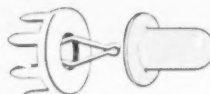
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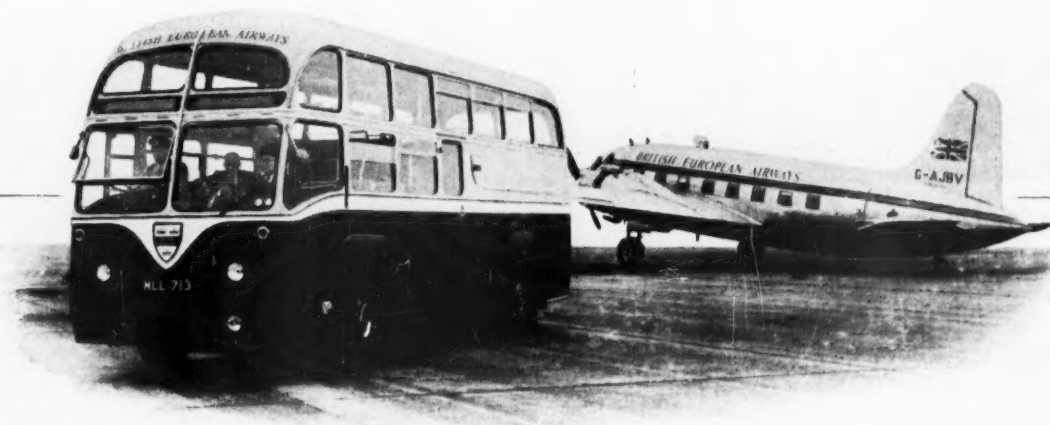
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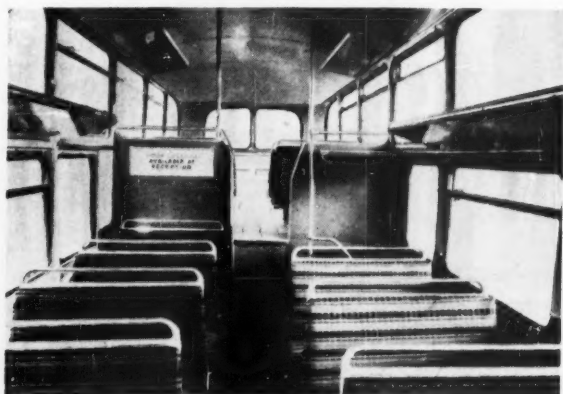


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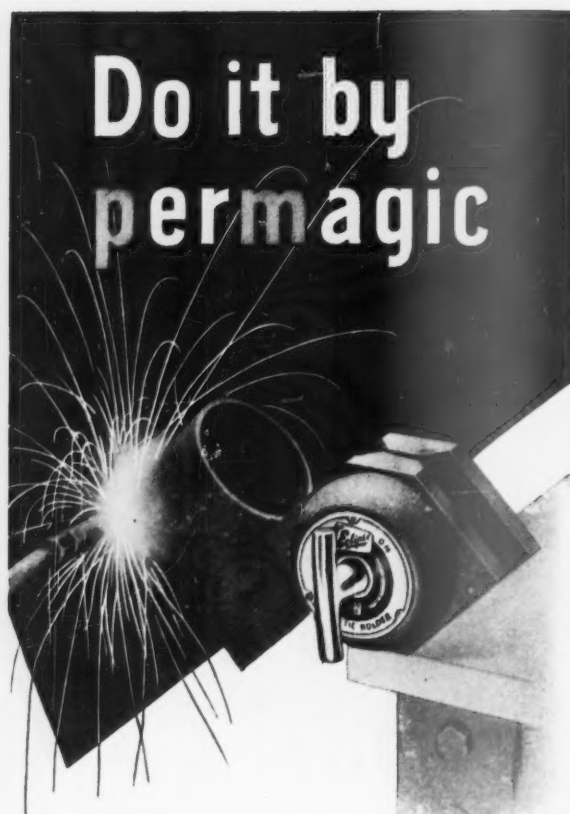
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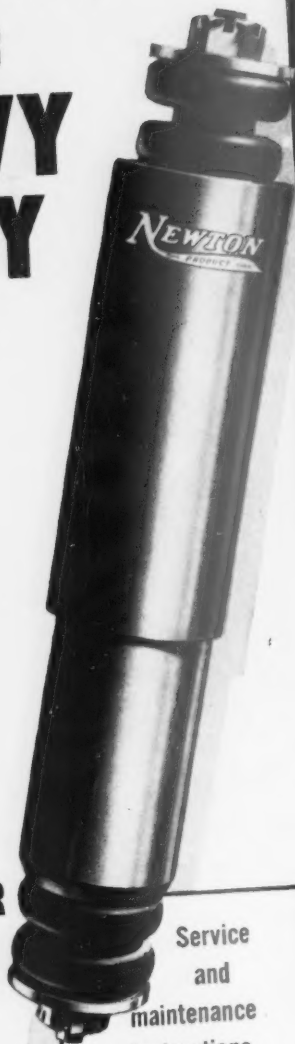
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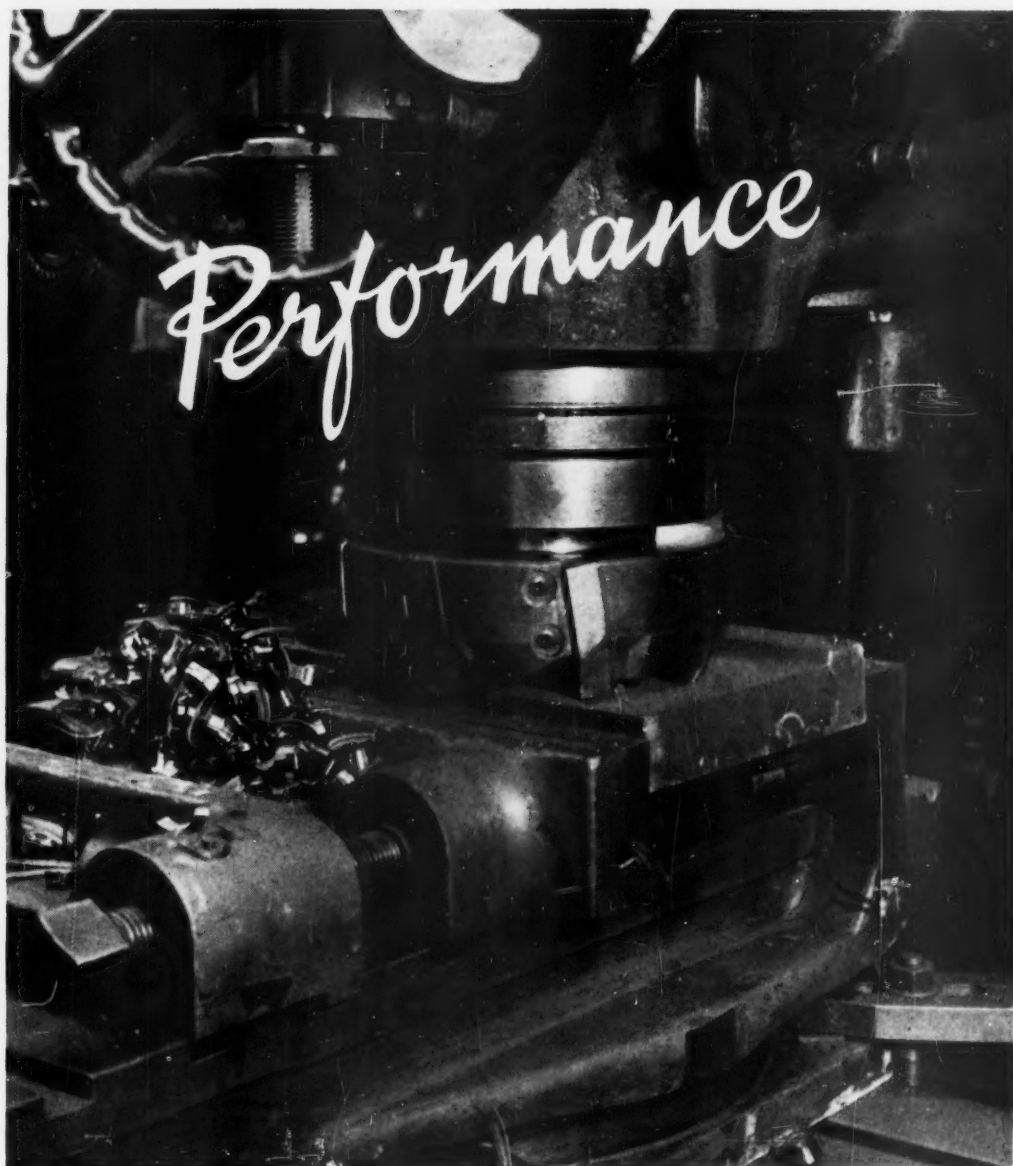
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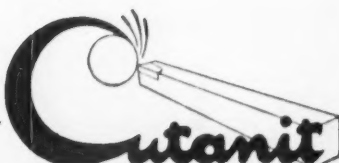
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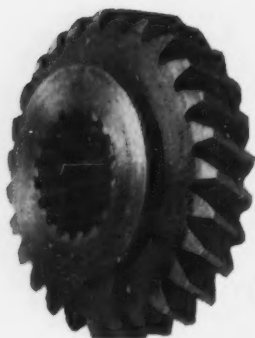
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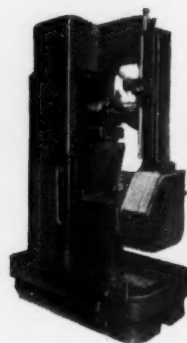
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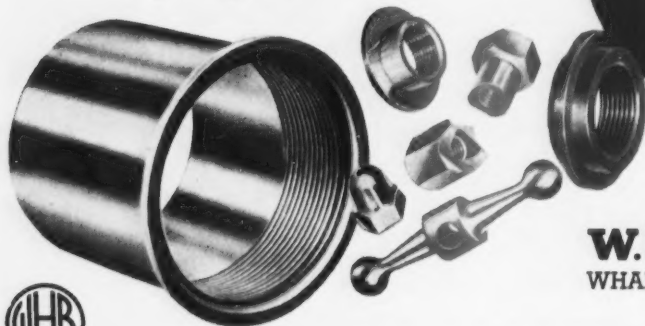
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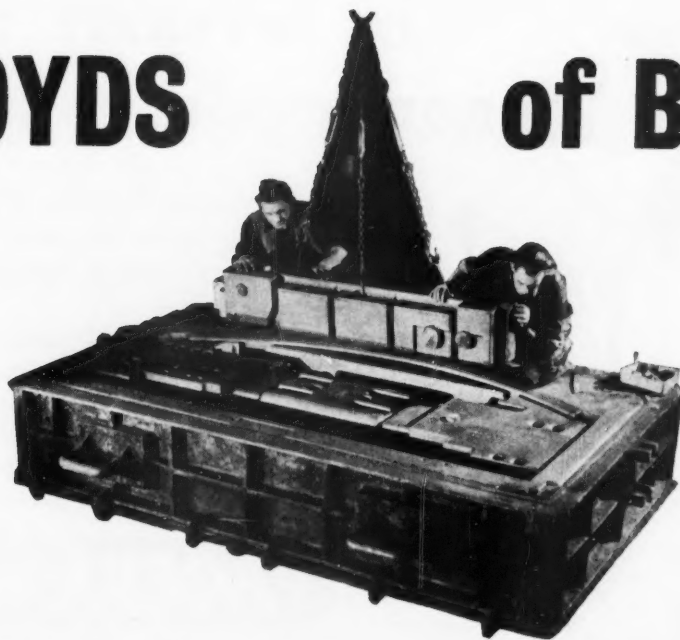
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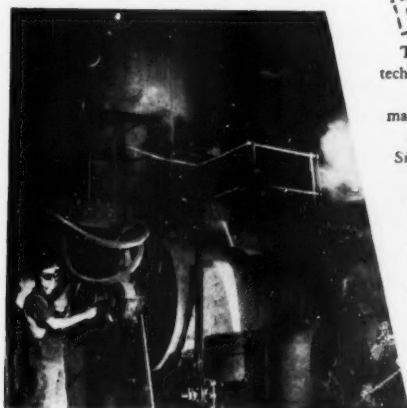
of Burton



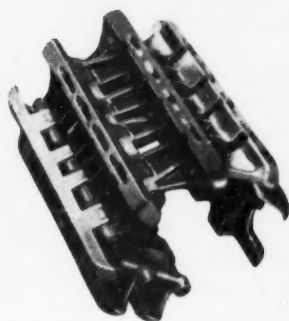
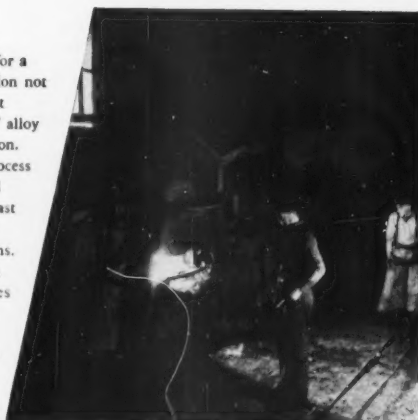
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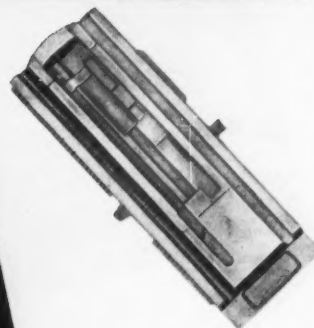
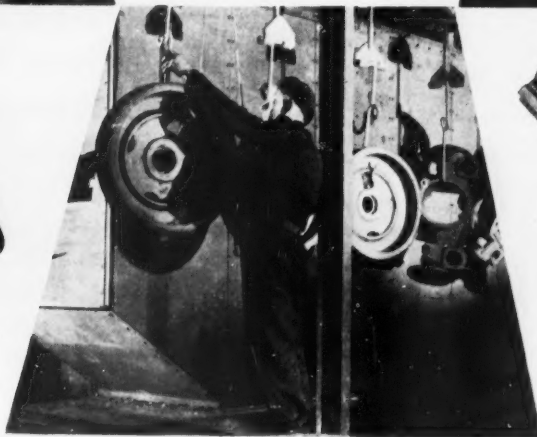
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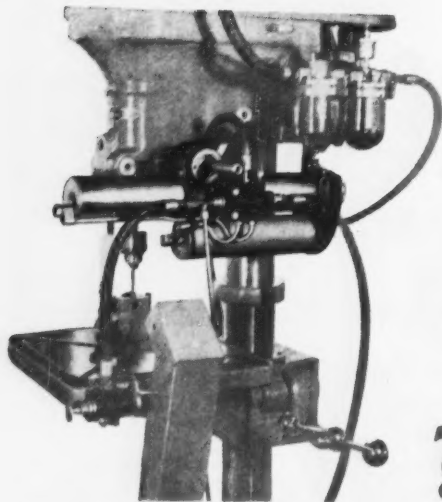
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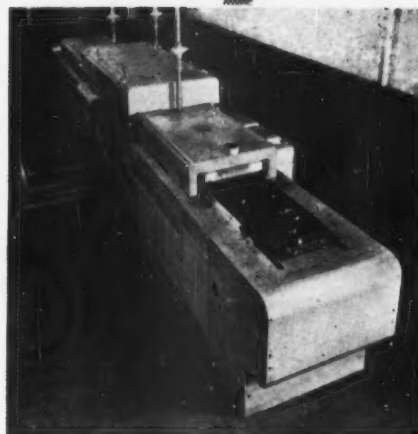
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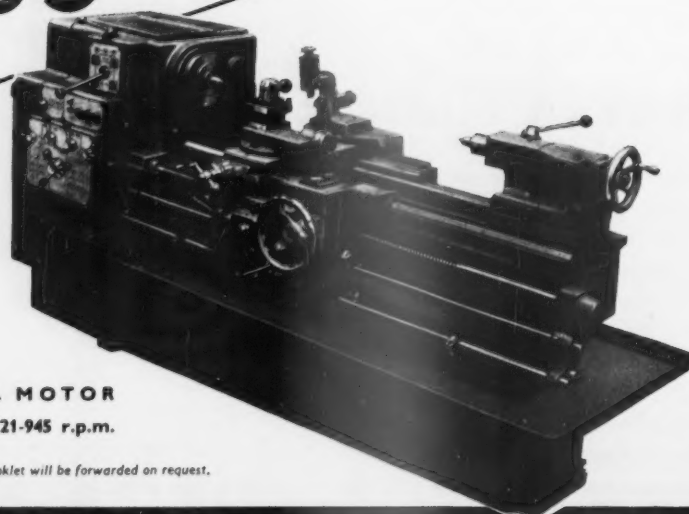
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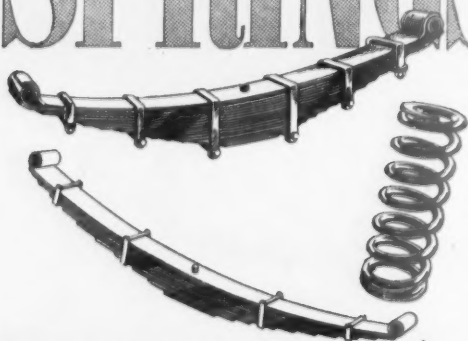
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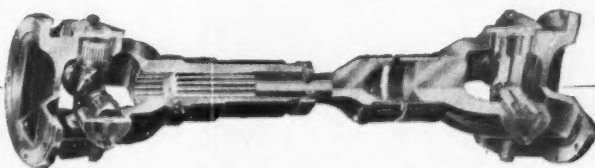


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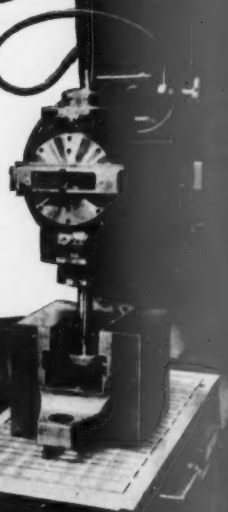
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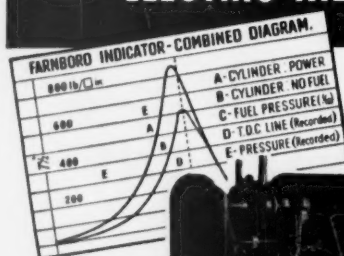


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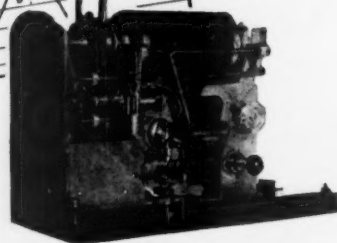
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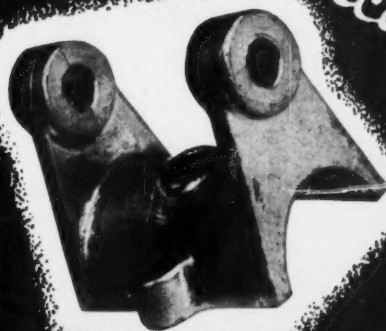


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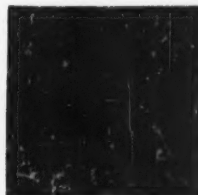
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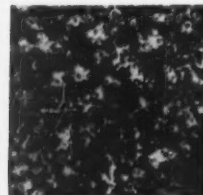
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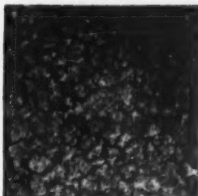
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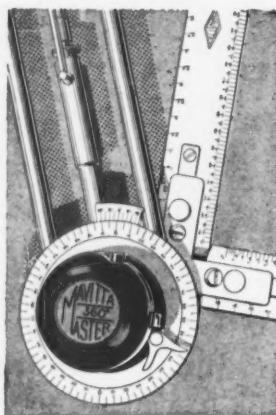
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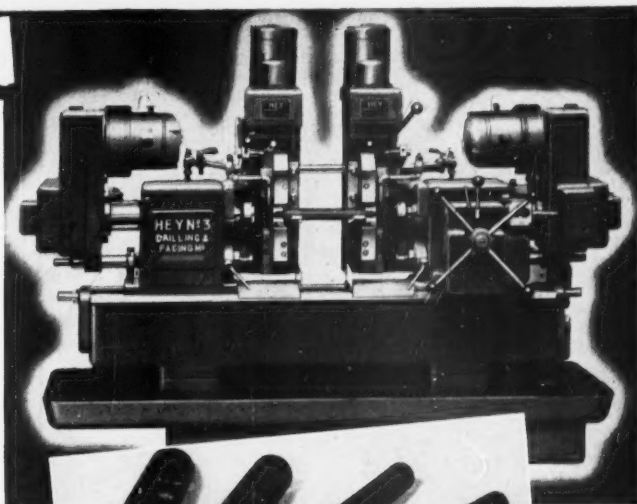
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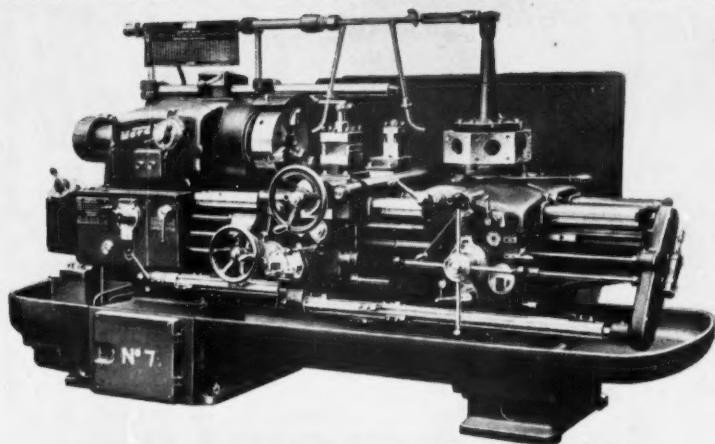
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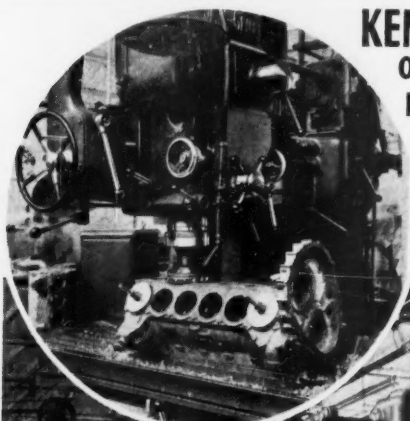
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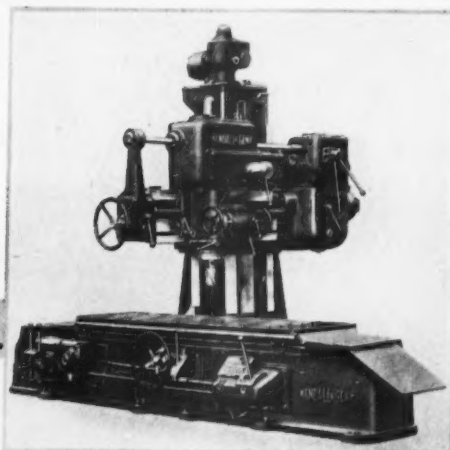
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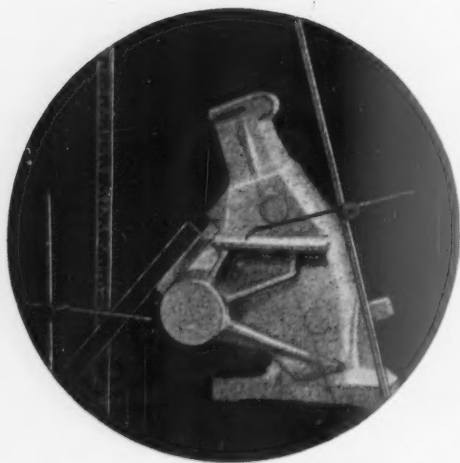
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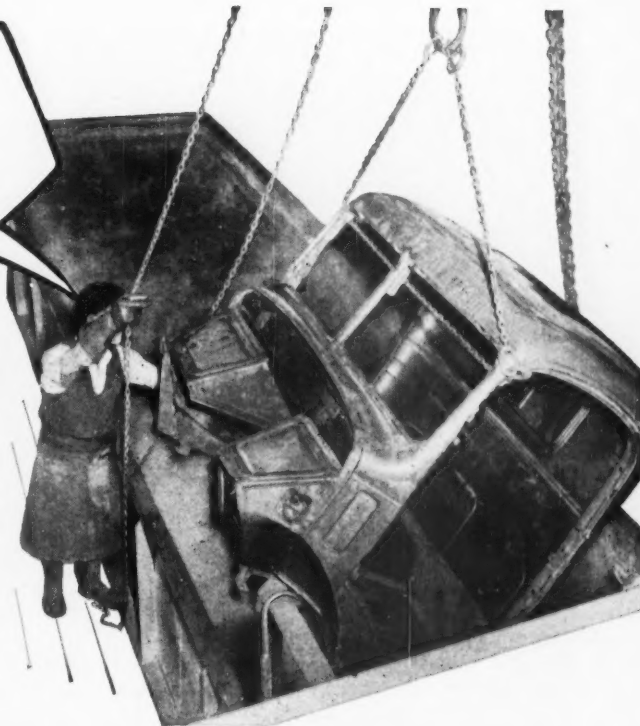


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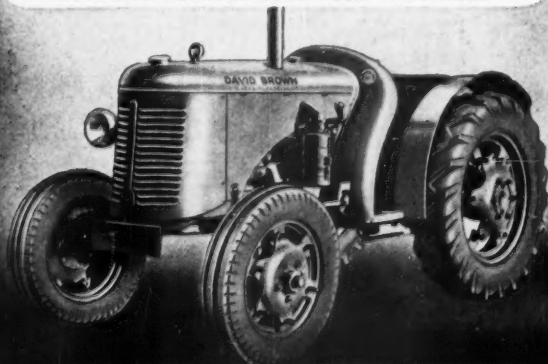
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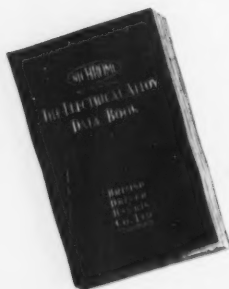
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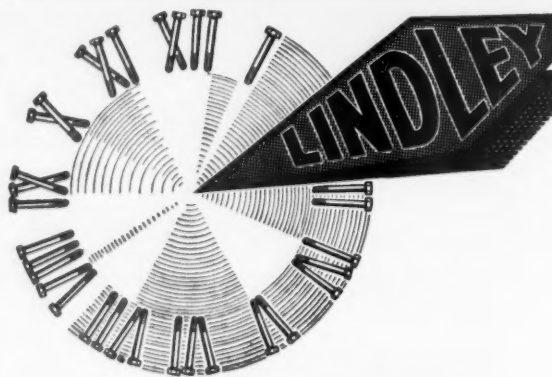
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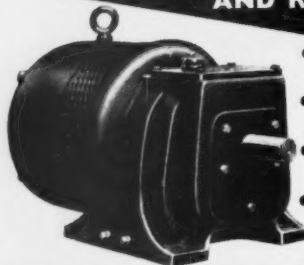
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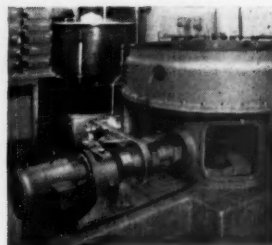
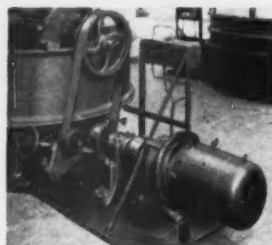
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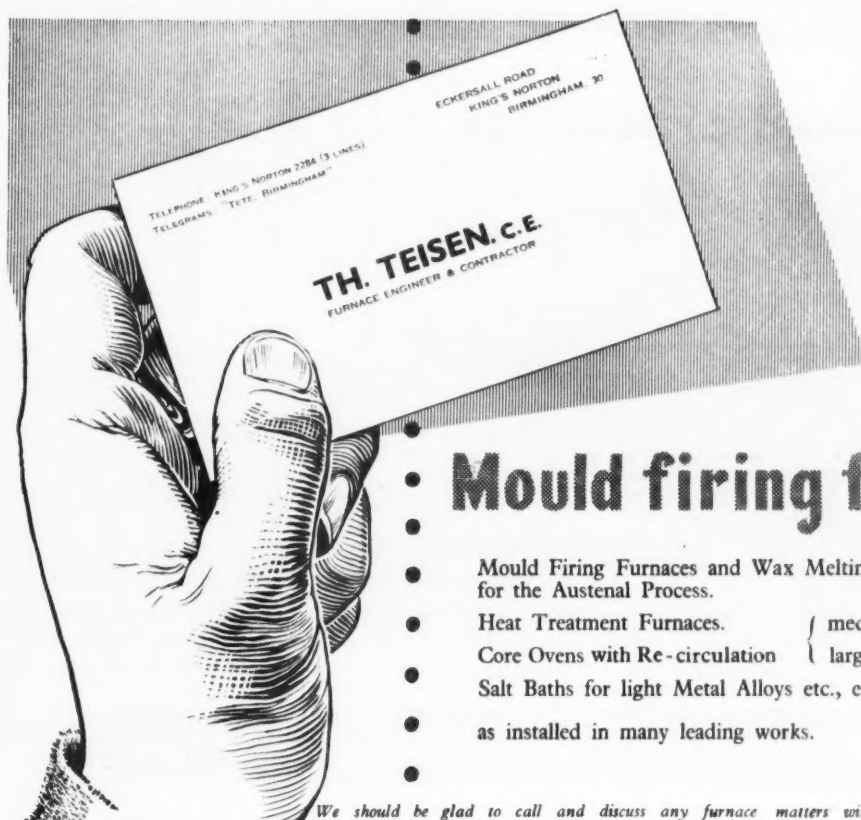
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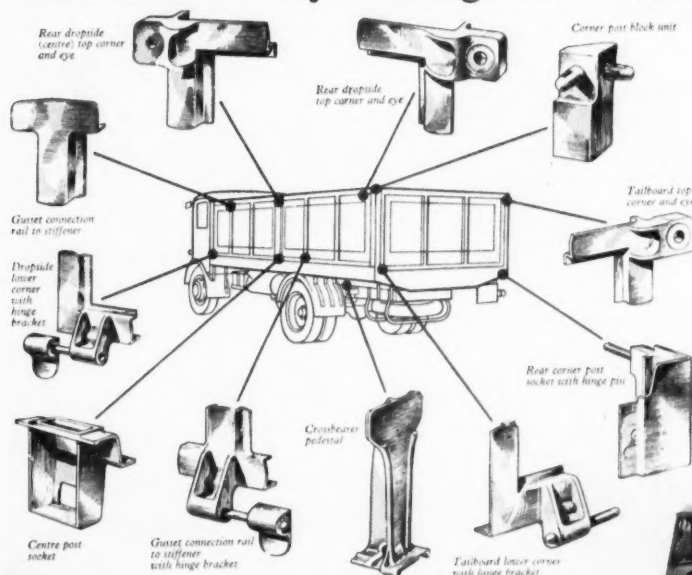


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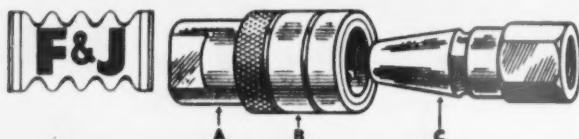
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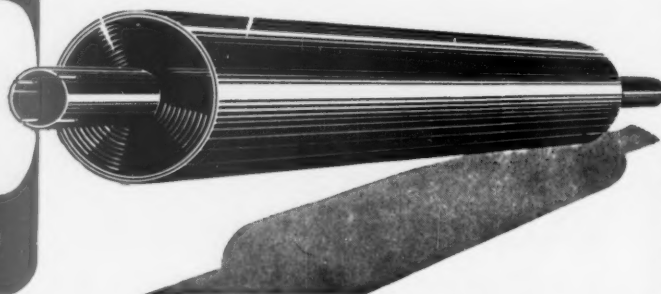
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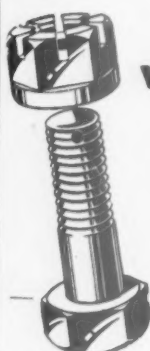
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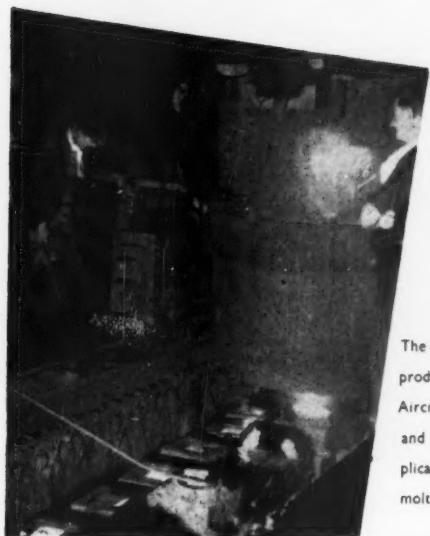
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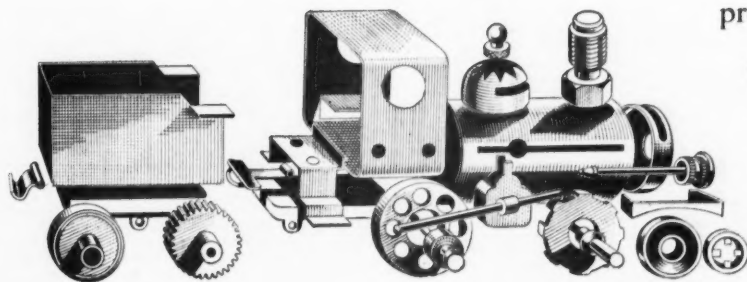
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